

CR151599

REPRESENTATIVE SHUTTLE

EVAPORATIVE HEAT SINK

Final Report No. T157-77

Contract No. NAS 9-14446

(NASA-CR-151599) REPRESENTATIVE SHUTTLE	N78-15160
EVAPORATIVE HEAT SINK Final Report (Vought	
Corp., Dallas, Tex.) 66 p HC A04/MF A01	
CSCS 20D	Unclas
G3/16	57823

Submitted by

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DALLAS, TEXAS 75222

to

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
JOHNSON SPACE CENTER
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FLASH EVAPORATOR FINAL REPORT

1.0 SUMMARY

The design, fabrication, and testing of a Representative Shuttle Evaporative Heat Sink (RSEHS) system which vaporizes an expendable fluid to provide cooling for the Shuttle heat transport fluid loop is reported. The RSEHS design evolved from previous flash evaporator development efforts to provide a minimum weight unit consistent with performance, structural, and vehicle interface requirements. The work was performed under NASA/JSC contract NAS9-14446 by the Vought Corporation. During the performance of this contract, the Shuttle Flash Evaporator System (FES) was awarded to another contractor. At NASA's direction work was stopped on development of the RSEHS with this report documenting the effort completed at the time of termination.

The optimized RSEHS minimum weight design was completed during the contract and meets or exceeds the Shuttle FES requirements. The Shuttle requirements and RSEHS design parameters are compared in Table 1-1. The RSEHS design is shown pictorially in Figure 1-1.

The RSEHS design approach uses two stages of cooling in each Freon loop. Trade studies have shown that the two-stage evaporator approach is the most weight effective approach for the Orbiter application, although the number of dynamic components is increased. On demand from the controller, water is allowed through the normally-closed metering valve and the nozzle which sprays the water onto the compact heat exchanger core surface. The controller senses Freon inlet temperature to the unit to determine if the unit should be active, and it monitors the Freon outlet temperature from each stage to determine if spray is required. Control strategy is to vary the controller signal pulse-width at a rate proportional to the magnitude of the sensed temperature deviation of the Freon outlet temperature from the set-point. Full redundancy is supplied to each spray chamber through a backup water supply, isolation valve, pressure regulator, metering valve, spray nozzle, and controller. Failure detection circuits are provided in each of the primary and backup controllers which provide a signal to the cabin in event of malfunction. The isolation valves are manually activated to switch from the primary to secondary systems.

Fabrication of half of the heat exchanger surfaces, all the metering valves, and the low temperature unit exhaust duct were completed prior to work stoppage. Checkout testing was being performed on the heat exchanger surfaces using simulated fluid manifolds at the time of work stoppage with the results verifying the flow/pressure drop integrity. Demonstration testing of

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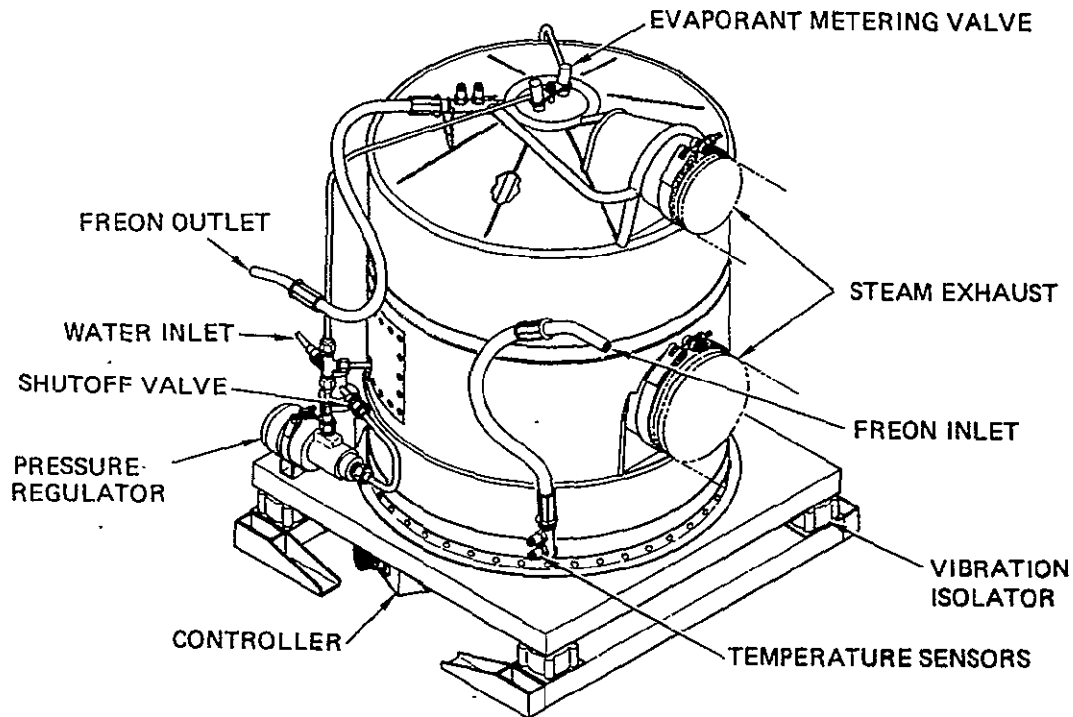


FIGURE 1-1 FLASH EVAPORATOR SUBSYSTEM

TABLE 1-1 FLASH EVAPORATOR PERFORMANCE COMPARISON

PERFORMANCE PARAMETER	HEAT REJECTION & OUTLET TEMP AT MAX FLOW RATE OF 2750 gph		PRESSURE DROP AT DESIGN FLOW RATE OF 2400 gph (PSI)	CONT POWER (WATTS)	WEIGHT		ENVELOPE		SUPERSONIC NOZZLE	
	(BTU/HR)	(°F)			FE INSTALL (LB)	SUPERSONIC NOZZLE INSTALL- LATION (LB)	FE INSTALLA- TION (IN x IN x IN)	SUPERSONIC NOZZLE INSTALLA- TION (IN x IN)	EXIT MACH NO	THRUST IMBALANCE (LB)
REQUIREMENT	131 000	37 ± 2	4.5	NOT SPECI- FIED	NOT SPECI- FIED	NOT SPECIFIED	18 x 18 x 28 (5.25 FT ³)	7" DIA x 10.5" LONG	3.0	≤ 0.008
DESIGN	131,000	37 ± 2	3.0	10 NOM 20 MAX	62.6 77.6 (WET)	11.6	15" DIA x 20" HIGH CYLINDER ON AN 18" x 18" x 3" BASE (2.61 FT ³)	7" DIA x 10.4" LONG	3.25	≤ 0.008

the heat exchanger surfaces using unheated valve/nozzle mounting plates, with no evaporator backcovers or exhaust elbow duct entry pieces was performed per NASA request to show the capabilities of the units to meet the Shuttle performance requirements with no ice formation on the evaporation surfaces. The units successfully operated with high evaporation efficiencies (91-1/2% to 96-1/2%) at both maximum and partial heat loads.

In addition to the RSEHS, a cold trap was also designed and fabricated to aid in testing the Shuttle FES in the NASA/JSC Crew Systems Division (CSD) test facilities. The cold trap will cryo-pump flash evaporator exhaust water from the CSD vacuum chamber test facility to prevent water contamination of the chamber pumping equipment. The cold trap consists of two standard large diameter pipes containing cold plate extrusions through which LN₂ is flowed to condense the FES exhaust water vapor. The two separate sections are required so that while one section is operating, the other section can be de-iced. This provides for continuous operation so that Shuttle missions can be simulated.

Detailed discussion of the RSEHS and cold trap design, fabrication and testing are contained in the sections that follow.

2.0 INTRODUCTION

The flash evaporator is a unique expendable heat sink device which absorbs heat from a coolant loop by vaporizing expendable water in a low pressure environment as depicted in Figure 2-1. Water is sprayed into a chamber formed by heat exchanger surfaces which contain the flowing liquid transport coolant (Freon). The water droplets strike the heat exchanger surfaces, and rapidly spread over the surface; the water is then vaporized as it absorbs heat from the coolant flowing in the heat exchanger. Evaporation of the water occurs so quickly that the heat exchanger surface appears to be dry; thus providing the name "dry-wall" evaporation for the process.

An exit duct, which is provided to exhaust steam from the spray chamber, is sized so that pressure in the spray chamber remains below the triple point of water (4.6 mmHg). The exit duct orifice diameter is fixed since no back pressure control is required at part load operation. No wicks, porous plates, or other special surface characteristics are required on the spray chamber walls.

Approximately 5% of the water spray rebounds from the spray chamber walls, and is entrained by the exiting steam. An elbow separator/heater is provided at the entrance of the exhaust duct to complete vaporization of the liquid water. The elbow separator is heated by the exiting Freon thus improving efficiency of the device and minimizing duct heating requirements.

Control is achieved by "on-off" modulation of the water metering valve as required to maintain the outlet Freon temperature at the desired level. When the Freon temperature is above the set-point, the "on" pulse width (and thus the fraction of "on" time) increases at a rate proportional to the sensed error. The same happens in reverse when the sensed error is below the control temperature, causing the "on" pulse width to decrease at a rate proportional to the sensed error. The total width of the pulse cycle ("on" time plus "off" time) is constant at 6 seconds. At full load spraying is continuous while at low load the water spray may be on for only a fraction of a second pulse every 6 seconds.

The RSEHS system was to be the final stage in the development of the flash evaporator system prior to its incorporation into the Shuttle Orbiter Active Thermal Control System. The flash evaporator concept has been under development by the Vought Corporation since the concept was successfully demonstrated in 1970. The Prototype 2 and 3 flash evaporators were developed in building block fashion with superior performance and installation flexibility to meet the wide range of evolving Space Shuttle requirements. The RSEHS program reported herein was to provide an optimized minimum weight flash evaporator system consistent with structural integrity which could meet the emerging performance and installation requirements of the Shuttle Orbiter. The work was performed for the Crew Systems Division of NASA-JSC under direction of Mr. Frank Collier.

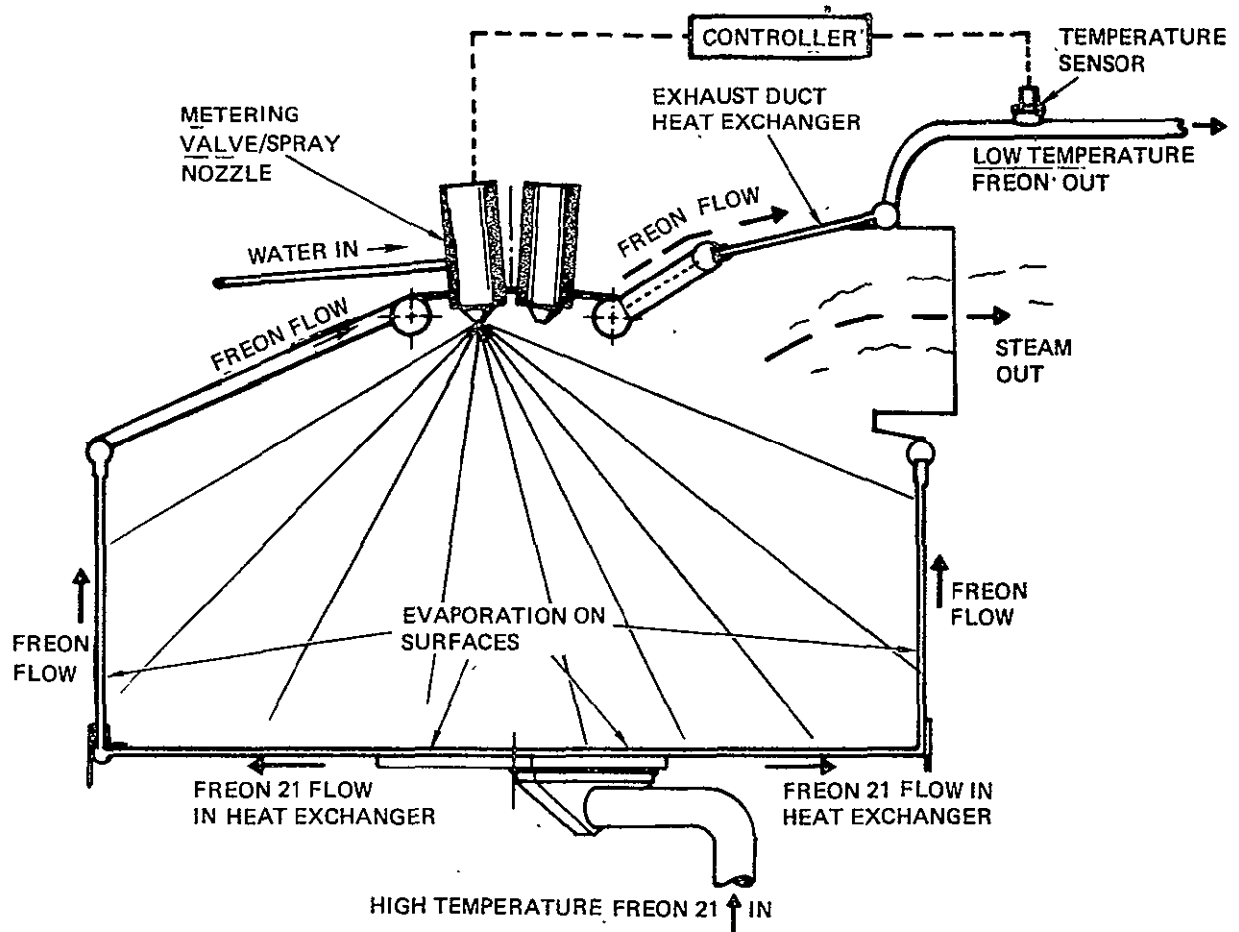


FIGURE 2-1: FLASH EVAPORATOR OPERATING CONCEPT

3.0 RSEHS DESIGN

The design requirements, design approach, and description of the RSEHS are presented in this Section.

3.1 Design Requirements

The Shuttle FES heat rejection requirements are shown in Table 3-1

TABLE 3-1 FLASH EVAPORATOR PERFORMANCE REQUIREMENTS

OPERATING MODE	EVAPORATOR HEAT LOAD (BTU/HR)	EVAPORATOR INLET TEMP (°F)	EVAPORATOR OUTLET TEMP (°F)	FREON 21 FLOWRATE (LB/HR PER LOOP)
ON-ORBIT (RADIATOR TOPPING)	3750 TO 20,000	40 TO 53	37 ± 2	2250 TO 2500
ON-ORBIT (EXCESS WATER DUMP)	30,000	60	37 ± 2	2250 TO 2500
ASCENT/ENTRY	131,000	132	37 ± 2	2750

Operationally the flash evaporator must respond instantly to maintain the proper Freon loop temperature following sustained dormant periods.

During on-orbit operation the flash evaporator must be activated when the Freon inlet temperature rises above 41°F, and is de-activated when the inlet temperature falls below 40°F. During on-orbit operation, all steam generated must be routed overboard through vents on opposite sides of the vehicle to nullify the thrust vectors on the Orbiter. Supersonic nozzles must be used on the vents to reduce water vapor contamination of the vehicle by accelerating the exhaust gas velocity to Mach 3.0. In ascent and re-entry modes of operation steam may be vented through a single duct.

The on-orbit water dump requirement is designed to boil off excess water produced by the Shuttle Orbiter fuel cells. No action on the part of the FES should be required to accomplish this: it is done by raising the radiator Freon outlet temperature set-point. The FES must then routinely evaporate water at the required rate.

The life design goals are:

- (1) Operating Life: perform all operations for a minimum of 10-years
- (2) Useful Life: operate for 20,000 hours, which is the equivalent of 100 orbital missions of 7-day length; the design shall not preclude extension of orbital stay time up to 30-days.
- (3) Shelf Life: up to 10-years from date of delivery

The functional reliability requirements of the FES were that it must be designed to meet single loop abort requirements following any single dynamic or structural component failure. Single loop abort requirements are defined such that the operating conditions, (e.g., flowrate, inlet temperature, and outlet temperature), of the active FES are the same as they were during normal operation as defined in Table 3-1.

Redundancy must be provided to dynamic components such that a first unknown failure followed by a second known or unknown failure will not lead to loss of crew or vehicle. Heat exchangers and pressure vessels are considered as structure for redundancy requirements.

3.2 Design Description

The RSEHS, designed to meet the above performance requirements, is shown schematically in Figure 3-1. The RSEHS flash evaporator (FE) assembly in each loop has been divided into two stages to improve performance. The low temperature stage provides on-orbit cooling. Non-propulsive steam ducts, with supersonic nozzles for contamination control, are connected only to the low temperature stage; the high temperature stage is connected to a single exhaust duct. When the FES is fully operational, i.e., the high temperature stage is active, there is no requirement for thrust nullification or contamination control.

An individual RSEHS FE assembly schematic is shown in Figure 3-2. Operation of the device is as follows: The temperature of the in-coming Freon is monitored to determine if the FES should be on or off, depending on whether the temperature is below 40°F (switch off) or above 41°F (switch on). Between 40 and 41°F it remains in whichever mode it is currently in. If the unit is active the controller causes water to spray into the high temperature stage unit to provide an outlet temperature of 60°F. The controller also monitors the low temperature stage outlet temperature and provides water spray to maintain the outlet temperature at $37 \pm 2^\circ\text{F}$.

Two failure detection circuits are provided in the controller. In the event that the unit should be functioning and is not, based on temperature readings, a warning light in the cabin is activated. The crew must take action to switch to the secondary water system, with its separate chain of electronics, pressure regulator, in-flow valve, and nozzle.

A flash evaporator assembly (two are required per shipset) is shown in Figure 3-3. The FE assembly fits within an allowable envelope of 18" x 18" x 28-1/4". The flash evaporators are stacked with the high-temperature unit on the bottom and the low-temperature unit on the top. This provides for an efficient Freon flow routing through the assembly since flow is generally in one direction from bottom to top, as shown in Figure 3-4. The Freon 21 flow enters the heat exchanger assembly at the center of the bottom of the high temperature unit. It flows through the heat exchanger core in the high temperature unit bottom and side walls, flows through the back cone transfer tubes, and vapor exit port heat exchanger. The flow then proceeds through the low temperature unit in the same manner as described above. Thus, the Freon flows in series through the high temperature unit first then through the low temperature unit.

The Shuttle Orbiter's primary and secondary water supplies are connected to each FE assembly. Water from each loop enters through the isolation valve and proceeds to the pressure regulator where the pressure is reduced from the 120 to 15 psia range down to $15 \pm 1/2$ psia. The water then flows to the metering valve where it is released to the spray nozzle on command from the controller. The spray nozzles are aimed at the center of the bottomplate (See Figure 2-1). Directing more evaporant on the hottest surfaces reduces surface area requirements for a constant sink capability. Stainless steel tubing, 3/16-in. diameter, is used in all water flow passages.

Redundancy is provided in the water flow system in each FE assembly. There are primary and secondary water metering valves and spray nozzles in each

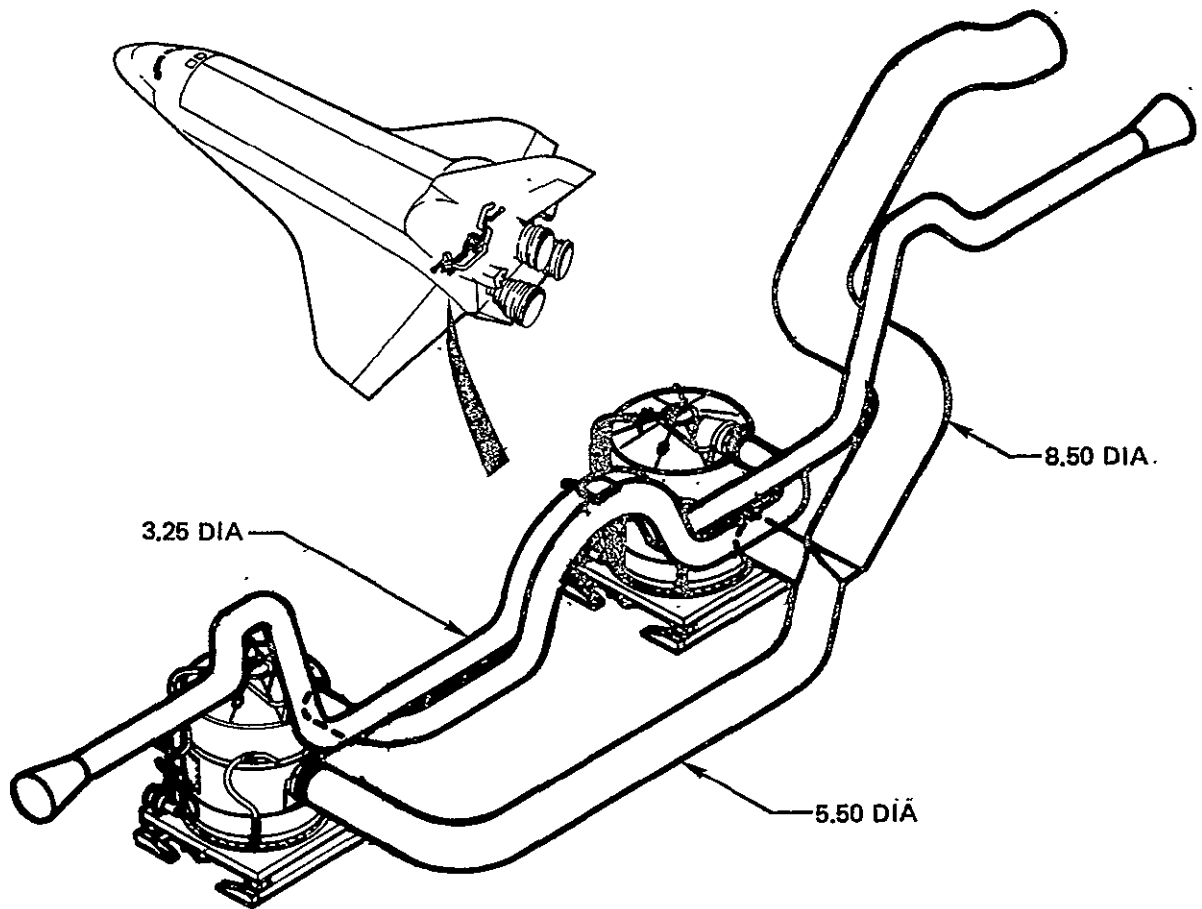


FIGURE 3-1 SHUTTLE INSTALLATION OF FES

evaporator spray chamber. Redundancy is also provided in the electronic controller so that separate circuits control the primary and secondary water loops. Temperature sensor redundancy is provided by duplicating sensor wells so that primary and secondary signals do not go through a single connector.

The FE assembly is shock mounted to reduce the vibration environment induced acceleration on components during launch. The design vibration spectrum is particularly severe on the FE assembly because it has a high energy level in the fundamental frequency, which is about 112 Hz. The valves in the FE experience loadings of over 100 g's rms if shock mounting is not used, with shock mounts the vibration loadings are below 6-g's rms. Displacement of the unit can be as much as 0.6-in. in the plane

parallel to the mounting bracket plane and 0.25-in. perpendicular to the mounting bracket plane. Provision is made on all lines and ducts to accommodate this motion. The water lines were designed with circular loops (or flexible hose) in them to allow this motion. Stainless steel bellows are provided on the ducts and special flexible hoses are used on the Freon lines. These flexible hoses serve as dissimilar metal isolators. The hoses are braided from polyamide with a Teflon liner, and with swaged-on ferrules of aluminum on one end and stainless steel on the other end. Teflon dissimilar metal isolators separate the aluminum heat exchanger and the stainless steel duct bellows.

The heat exchangers are an all aluminum, brazed and welded assembly. Materials used in manufacture of the heat exchanger are 6951 fins, No. 23 braze sheets, 6061 tubing and 6061 for the sheet metal cone and miscellaneous structural and mounting provisions. A corrosion preventative chemical coating per MIL-C-5541 A, Class 1A is applied to the heat exchanger units. The heat exchangers are permanently attached to an aluminum honeycomb mounting plate. The facesheets and hexagonal cell core pattern are adhesively bonded together. The honeycomb plate is attached to the vehicle structure through four elastomeric mounts. Pressure regulator and isolation valve assemblies are mounted on the honeycomb plate. The controller is mounted beneath the honeycomb plate; however, the connectors are placed so they can be mated and detached without removing any structure.

The RSEHS system weight is 28.4 Kg (62.6 lbs) dry plus 5.3 Kg (11.6 lbs) for the two supersonic exhaust nozzle installations. The total weight is 40.5 Kg (89.2 lbs) wet. The pressure drop for the FE assembly in each Freon loop is 20.7 KPa (3 psi) at a Freon flowrate of 1089 Kg/hr (2400 lb/hr). The heat rejection of the FES is 38.4 Kw (131,000 BTU/hr) with a Freon flowrate of 1247 Kg/hr (2750 lb/hr) in each Freon loop, and a Freon outlet temperature of $2.8^{\circ} \pm 1.1^{\circ}\text{C}$ ($37^{\circ} \pm 2^{\circ}\text{F}$).

3.3 Component Description

A detailed description of the RSEHS components including the performance and operational characteristics is presented in this section. Where possible, emphasis has been placed on obtaining components which have already been qualified to similar requirements to minimize the development costs to the program while reducing technical risk. The components are identified in Table 3-1, the schematic of Figure 3-2, and the design drawings of Figure 3-3.

FES HEAT EXCHANGER ASSEMBLY

The RSEHS heat exchanger assembly, shown in Figure 3-5, consists of two evaporative heat exchangers in a single assembly - one (the low-temperature unit) for the top-off and water dump modes and the other (high-temperature unit) for "full-up" mode. One FES shipset of hardware consists of two FES exchanger assemblies; one for each Freon loop. Each heat exchanger unit contains provisions for mounting the water metering valves near the apex of the cone where the liquid water enters in the form of a spray. The evaporant vapor leaves through an exit port provided on each heat exchanger unit.

As shown in Figures 3-4 and 3-5, Freon enters the high temperature unit in the center of the circular bottom plate heat exchanger, flows radially outward to the edge of the plate where it turns and flows up the cylindrical heat exchanger section into the manifold ring. The Freon flow then flows radially inward through tubes (which supply heat to the closure cone) to the doughnut manifold (which heats the valves and nozzles). Part of the Freon flow is then split off to flow through the water separator/evaporator elbow at the entrance to the exit duct. The Freon flow is then re-combined at the entrance to the low temperature stage. The Freon path through the low temperature stage is the same as through the high temperature stage. Details of the heat exchanger unit construction are shown in Figure 3-5a.

The heat exchanger cores are vacuum-fluxless brazed from 0.41 mm (0.016 in.) Number 23 braze sheet (6951 aluminum) and 0.13 mm (0.005 in.) thick 6951 aluminum fins which are 2.54 mm (0.10 in.) height. The cylindrical side walls and the bottom plate with inlet manifold are brazed separately and are heat treated to a T-6 condition. They are subsequently fusion welded together and a 6061-T6 aluminum manifold distribution ring which is 12.7 mm (0.5 in.) in diameter and 0.51 mm (0.020 in.) thickness is welded to the cylindrical section. Eight transfer tubes 9.5 mm (3/8 in.) diameter and 0.51 mm (0.020 in.) thickness are welded to the manifold distribution ring and to a 19.05 mm (3/4 in.) diameter Freon collector ring. The valve mounting plate, welded to the Freon collector ring, is also a vacuum-fluxless brazed heat exchanger core with mounting provision for the metering valve/nozzle assemblies.

The high and low temperature heat exchanger assemblies are permanently joined through a 0.51 mm (0.020 in.) 6061-T6 aluminum cylindrical section which has structural access doors, and which is fusion welded to each heat exchanger. Freon lines inside the assembly are 6061-T6 aluminum tubes, which are joined by fusion welding.

TABLE 3-2 MAJOR RSEHS COMPONENTS

ITEM	CONSTRUCTION	REMARKS
HEAT EXCHANGER	VACUUM-FLUXLESS BRAZED ALUMINUM CORE WITH FUSION WELDED MANIFOLDS.	STD HEAT EXCHANGER CORE CONFIGURED TO FORM SPRAY CHAMBER.
ELECTRONIC CONTROLLER	DISCRETE JANTX-V COMPONENTS ON PRINTED WIRING BOARDS IN AN ALUMINUM ENCLOSURE.	SIMILAR TO APOLLO HEATER CONTROLLER.
HIGH AND LOW TEMPERATURE STAGE SPRAY NOZZLES	STAINLESS STEEL BODY	SIMILAR TO JET ENGINE FUEL ATOMIZERS.
HIGH TEMPERATURE METERING VALVE	STAINLESS STEEL BODY WITH NEOPRENE O-RING SEALS.	USED ON THE NASA-GODDARD AEROBEE ROCKET ACS AND ON OSO PROJECT. APPROX. 500 HAVE BEEN DELIVERED. HAS BEEN DEMONSTRATED TO HAVE CYCLE LIFE OF OVER 250,000 THE VALVE WILL HAVE THE INLET PORT RELOCATED.
LOW TEMPERATURE METERING VALVE	STAINLESS STEEL BODY WITH NEOPRENE O-RING SEALS.	USED ON IMP SATELLITE
WATER PRESSURE REGULATOR	STAINLESS STEEL BODY WITH NEOPRENE O-RING SEALS.	SIMILAR TO UNIT USED IN 2000 HOUR GE FUEL CELL ENDURANCE TEST.
WATER ISOLATION VALVE	STAINLESS STEEL BODY WITH STAINLESS STEEL BELLOWS SEALS	HAS BEEN QUALIFIED FOR: INTELSAT IV, ATS F/G, AND CLASSIFIED AF VEHICLE. 100 HAVE BEEN BUILT; 30 ARE PERFORMING IN SYNCHRONOUS ORBIT NOW.
TEMPERATURE SENSOR	THERMISTOR IN PROBE WHICH CAN BE REMOVED WITHOUT BREAKING INTO FREON SYSTEM.	SIMILAR TO THOSE USED IN THE ORBITER FREON LOOPS
SUPERSONIC NOZZLE	FORGE SPUN FROM HAYNES ALLOY 188 AND MACHINED TO FINAL CONFIGURATION	AEROSPACE RCS MOTOR TECHNOLOGY UTILIZED

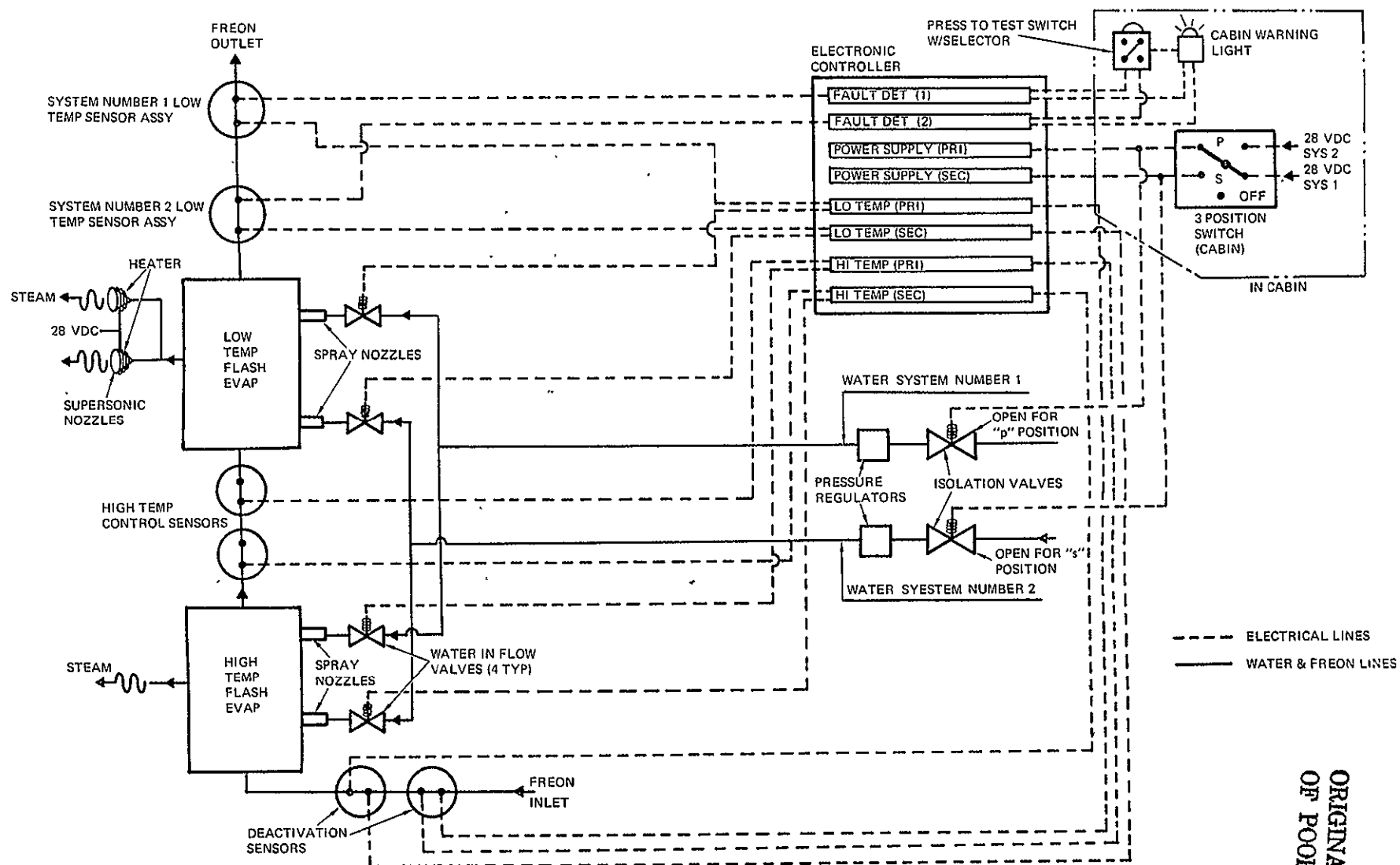
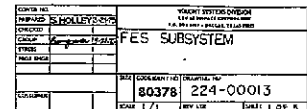


FIGURE 3-2 FLASH EVAPORATOR ASSEMBLY SCHEMATIC

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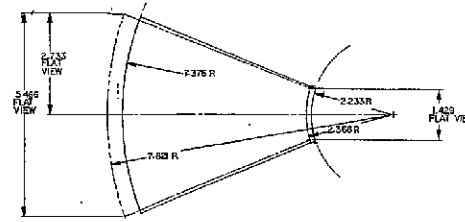
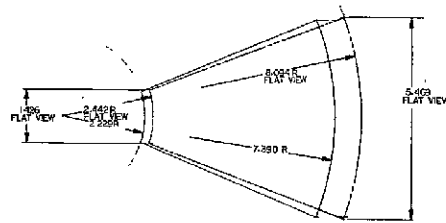
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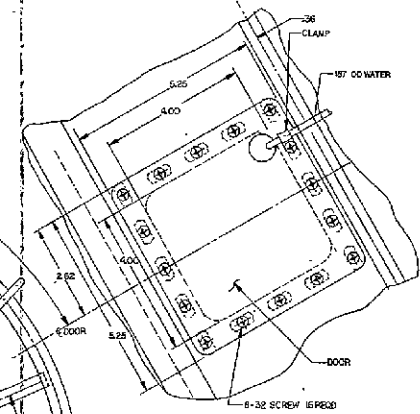
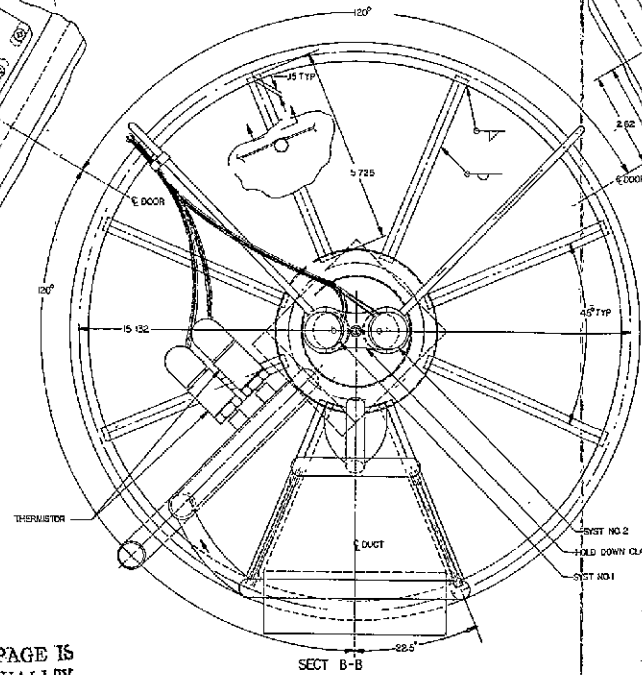
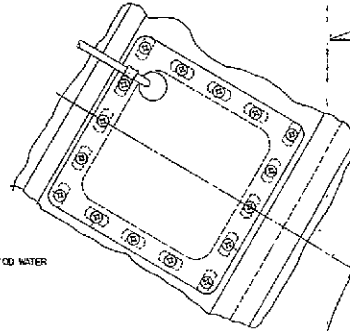
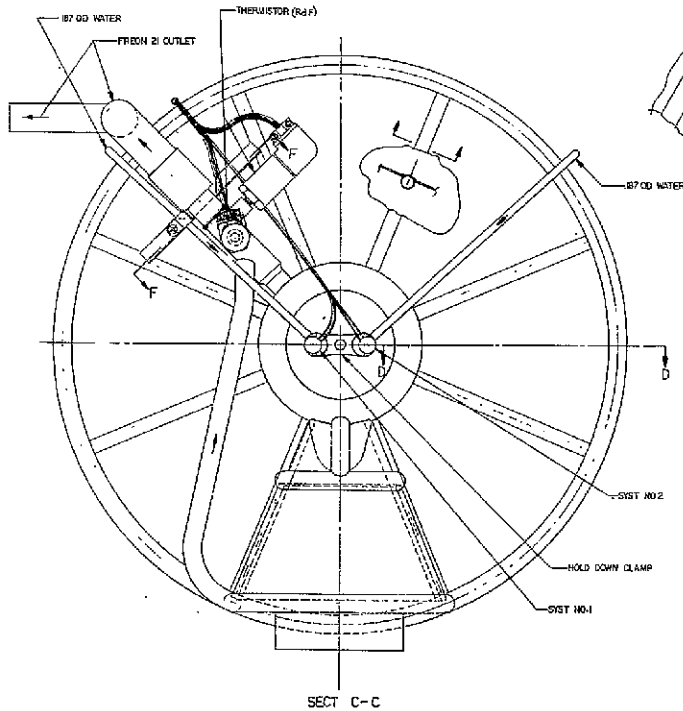
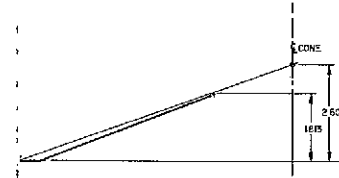
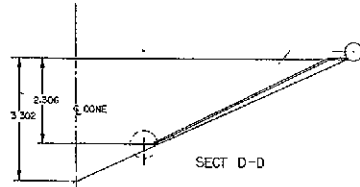


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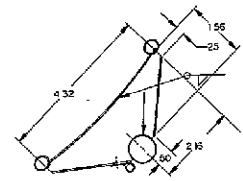
FIGURE 3-3, CONTINUED

CONTROL NO.	80378	UNIDENTIFIED SYSTEM
DESIGNER	SHAWLEY-BR	174-144444-1000000000
ENGINEER	SHAWLEY-BR	174-144444-1000000000
DRIVER	SHAWLEY-BR	174-144444-1000000000
TESTER	SHAWLEY-BR	174-144444-1000000000
REVIEWER	SHAWLEY-BR	174-144444-1000000000
DATE	10/1/71	174-144444-1000000000
SCALE	1/1	174-144444-1000000000
REV	1	174-144444-1000000000
DATE	10/1/71	174-144444-1000000000

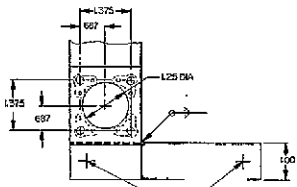
FOLDOUT FRAME

FOLDOUT FRAME 2 ORIGINAL PAGE IS OF POOR QUALITY

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SECT F-F



SECT B-B

WATER INLET
SYST NO. 2
INTERFACE

WATER INLET
SYST NO. 1
INTERFACE

2217 DIA-24 LATCHING
VALVE CARTRIDGE
SYST NO. 1

2755-0000 PRESSURE REG
CARTRIDGE
SYST NO. 1

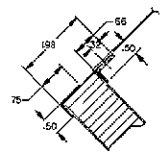
CONTROL BOX

16.37 DIA

SECT E-E

AFT

R-1 STRUCTURAL
INTERFACE TYP



SECT A-A

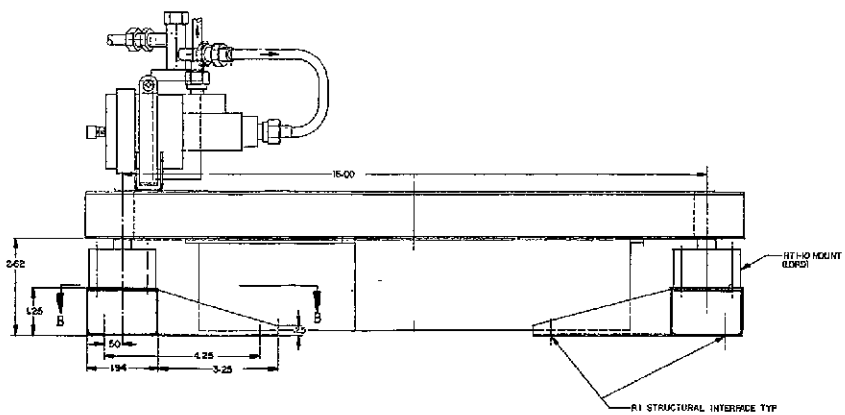


FIGURE 3-3, CONTINUED

CONTRACT NO.	80378	PROJECT NO.	224-00013
DESIGNED BY	W. KELLEY	CHECKED BY	W. KELLEY
DRAWN BY	W. KELLEY	DATE	1/1
REVISIONS			
DESCRIPTION	FES SUBSYSTEM		
DATE	1/1	REV	1

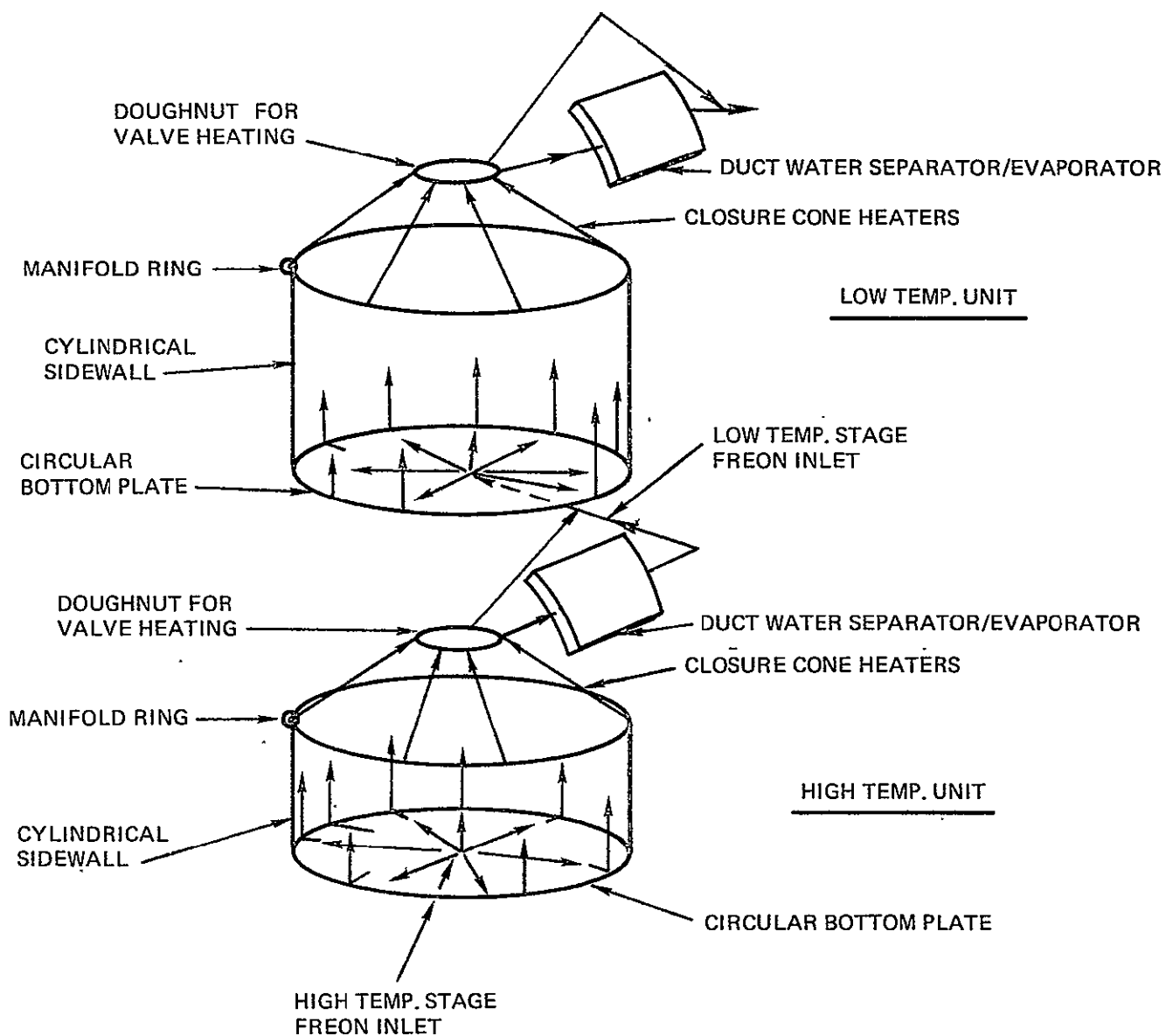


FIGURE 3-4 ROUTING OF FREON THROUGH HEAT EXCHANGER ASSEMBLY

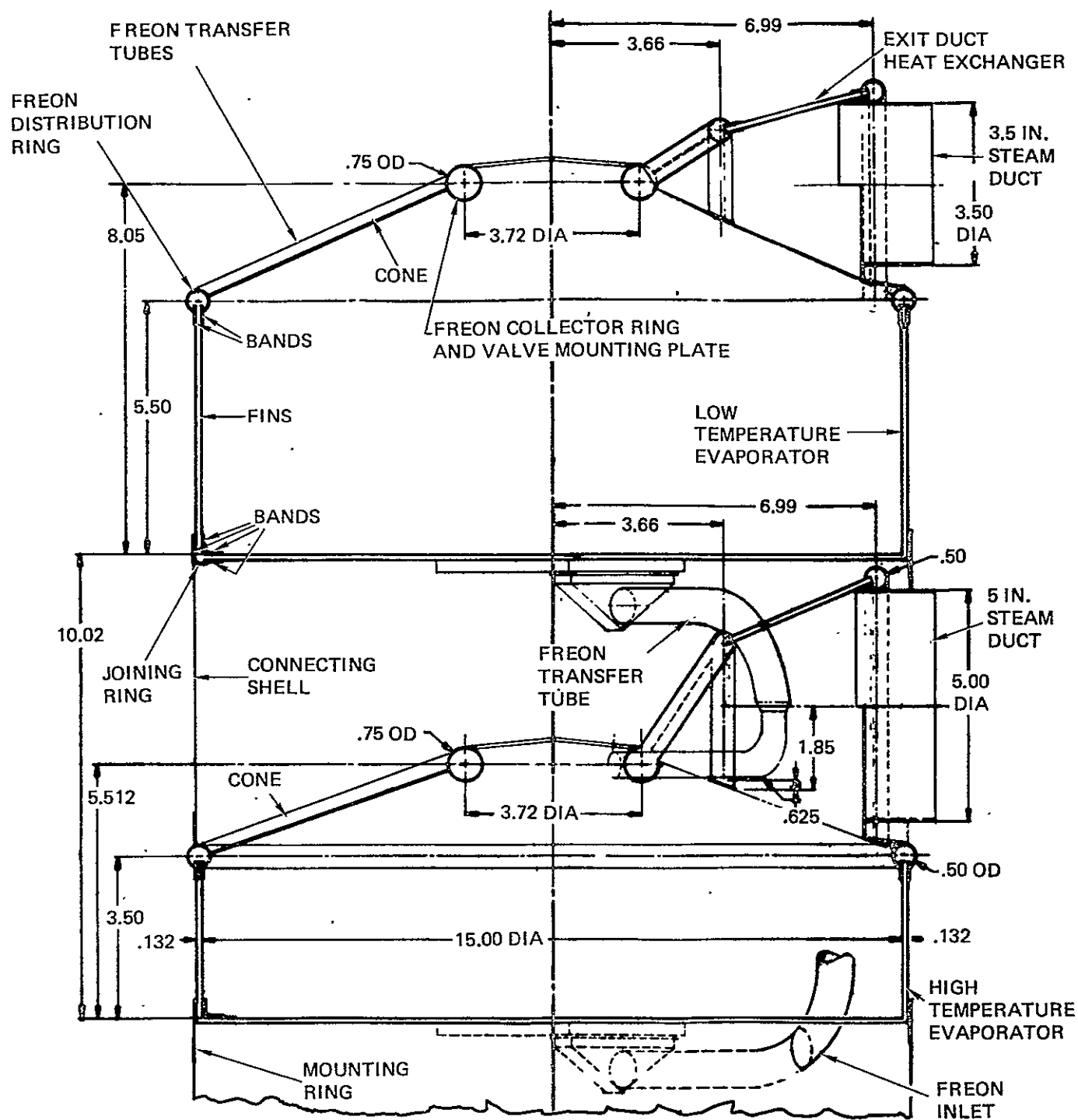


FIGURE 3-5 HEAT EXCHANGER ASSEMBLY CROSS SECTION

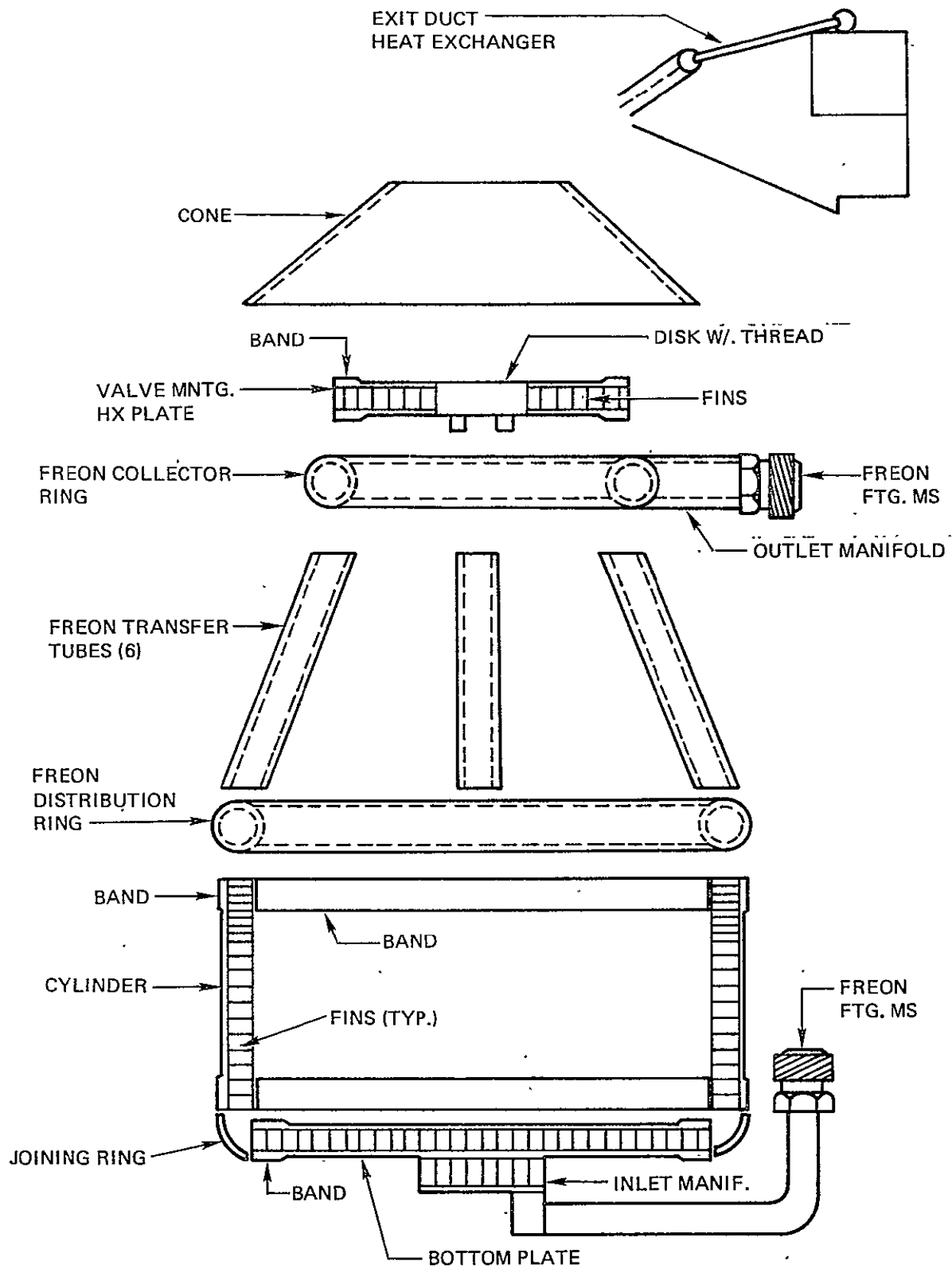


FIGURE 3-5a HEAT EXCHANGER ASSEMBLY COMPONENTS

The heat exchanger assembly is permanently attached to an aluminum honeycomb mounting plate. This plate is formed from 1.02 mm (0.040 in.) thick 2024-T81 face sheets and 5052-H39 core formed from 0.03 mm (0.001 in.) sheets in a 4.76 mm (3/16 in.) hexagonal cell pattern (density is 44.66 Kg/m² (3.1 lb/ft³)). Sheets and core are bonded in an autoclave using Metlbond 329-7 adhesive.

LOW TEMPERATURE METERING VALVE

The low temperature metering valve is a normally closed solenoid valve designed for fast response, high cycle life and minimum weight. It was previously used on the IMP Satellite. Valve shut-off is accomplished by a stainless steel plunger sealing against a molded elastomer seat (see Figure 3-6). In operation (with the valve open) water enters the valve and flows axially through the center of the bellows to the valve seat area. Flow travels between the plunger and seat and into the spray nozzle entrance. Volume downstream of the seat area has been minimized to prevent ice formation in residual water exposed to the vacuum following valve closing. Integration of the spray nozzle into the valve is also shown in Figure 3-6. A summary of the valve function, performance, and design is described in Figure 3-7.

HIGH TEMPERATURE METERING VALVE

The high temperature metering valve is a normally closed solenoid valve. It was previously used in the attitude control system of the NASA Goddard Aerobee Sounding Rocket. The valve was designed for fast responses, high reliability and high cycle life. Valve sealing is accomplished by a stainless valve head sealing against a Kel-F seat (see Figure 3-8). Dynamic sealing is accomplished with a welded metal bellows and static sealing is accomplished with neoprene o-ring seals.

Flow enters the valve radially near the valve seat and exits axially. At a flowrate of 24 Kg/hr (53 lb/hr) of water, the opened valve has a pressure drop of 2.1 KPa (0.3 psi). In the closed position, the valve leakage is less than 1 cc/hr of water at a pressure of 829 KPa (120 psi).

The valve is fabricated from stainless steel materials (303 and 430F) except for the Kel-F seat, the nickel bellows, and neoprene o-rings. The water spray nozzle is integrated into the valve as shown in Figure 3-8. The amount of volume downstream of the seat has been minimized to reduce the amount of liquid that evaporates after the valve is closed. A summary of the valve function, performance, usage history, modifications for the RSEHS program and design description are provided in Figure 3-9.

ISOLATION VALVE

The isolation valve is a latching solenoid which has been utilized successfully on three separate space programs. Latching is accomplished by the snap-through of a light weight, highly reliable belleville spring with two stable positions. This method was chosen over other latching methods such as magnetic latching, mechanical detent latching, and mechanical trip latch because of its high reliability, light weight, and long cycle life capabilities.

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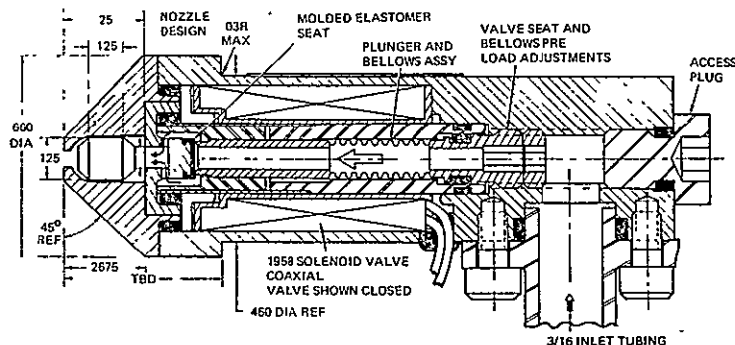


FIGURE 3-6 LOW TEMPERATURE METERING VALVE

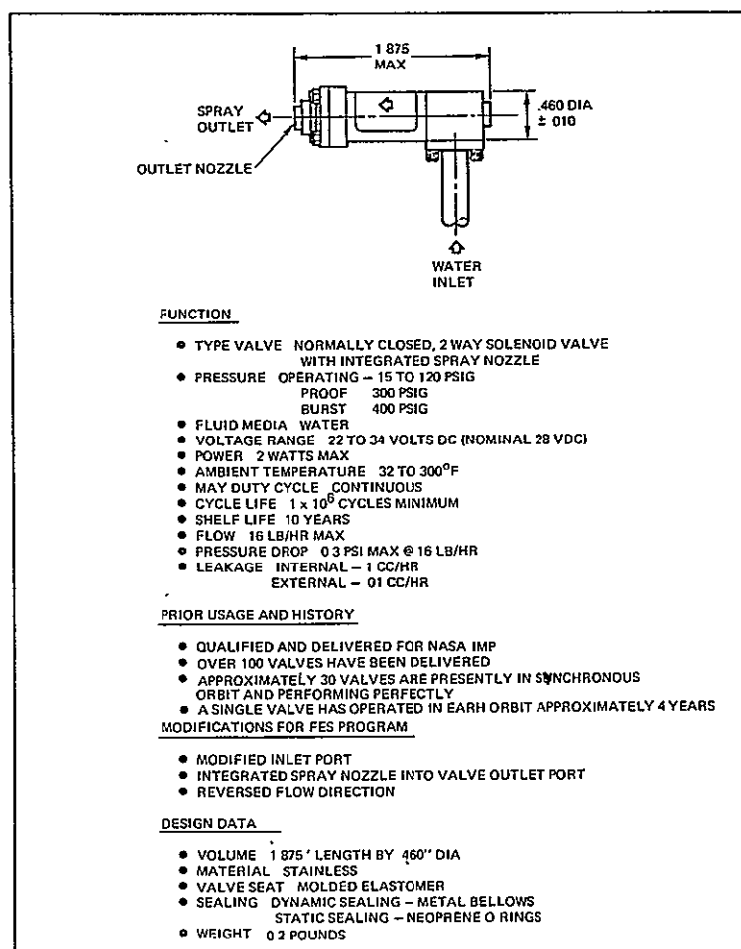


FIGURE 3-7 LOW TEMPERATURE METERING VALVE

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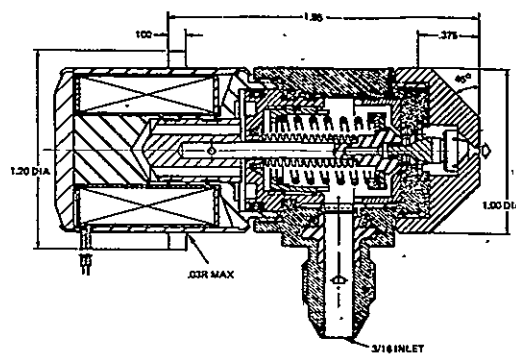


FIGURE 3-8 HIGH TEMPERATURE METERING VALVE

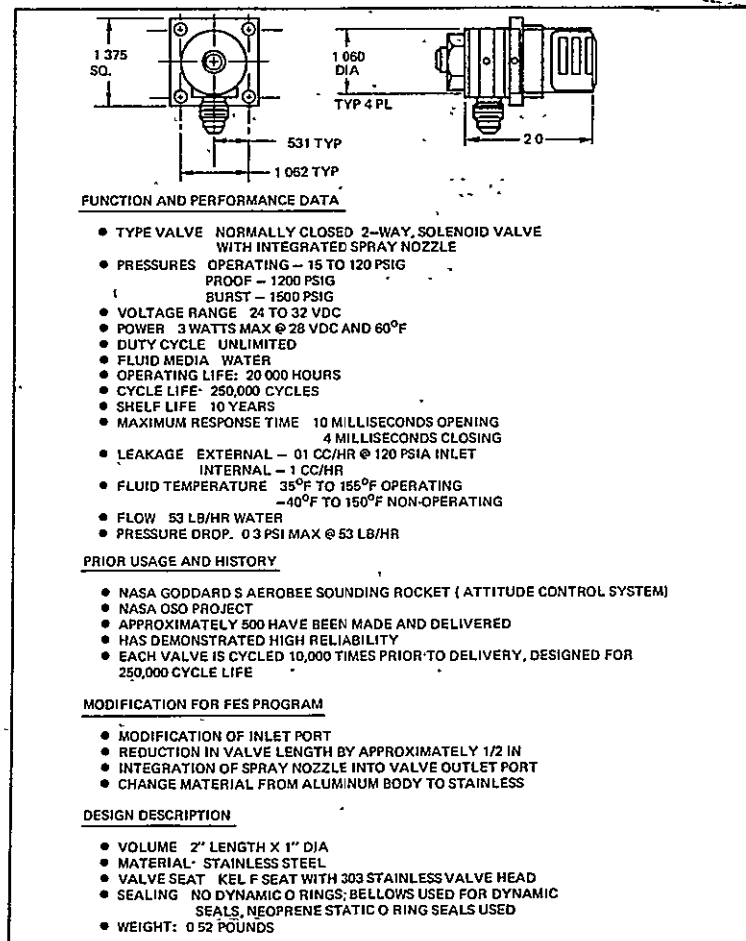


FIGURE 3-9 HIGH TEMPERATURE METERING VALVE

Flow enters and exits the valve sealing port in a radial direction (see Figure 3-10). The poppet head seals against a TFE Teflon seat which is completely enclosed except where contact is made with the metal sealing surface. The Teflon can therefore withstand relatively high compressive forces without cold flow. A thermal curing process is used to further guarantee against cold flow of the Teflon. The valve head is slightly unbalanced relative to the outlet pressure to permit the valve to relieve slightly back into the water systems at pressures of 1035 KPa (150 psi) over the inlet pressure to eliminate overpressure problems due to thermal expansion of water trapped in the line between the metering valve and the isolation valve for a dormant system. Pressure drop of the valve is less than 2.8 KPa (0.4 psi) at a water flowrate of 31.3 Kg/hr (69 lb/hr). Both static and dynamic sealing for the valve is accomplished by two metal bellows. One bellows seals inlet pressure from the ambient while the other bellows seals the outlet pressure from the ambient. Valve poppet and seat assemblies are the only valve components wetted by the fluid. The valve body is 300 series stainless. Figure 3-11 gives a summary of the valve performance data, prior usage and design description.

WATER PRESSURE REGULATOR

The water pressure regulator is an inlet pressure balanced design with a molded rubber seat. It functions as follows (see Figure 3-12). Water enters the unit from the bottom and proceeds upward towards the valve head and seat. The valve head is attached to a bellows in a way that makes it inlet pressure balanced. This means that variations of inlet pressure do not create force variations inside the regulator which will cause changes to the regulated outlet pressure.

The gage outlet pressure is sensed by the aneroid. As the outlet pressure begins to decrease, the bottom plate of the aneroid begins to move in the downward direction. This is because the combined force of the helical spring and the Belleville spring is now greater than the decreasing force of the outlet pressure acting against the aneroid plate. A set of pins transmits the downward motion of the aneroid to the valve head, thus causing it to open and allow flow.

When the flow is greater than the downstream demand, the pressure at the outlet of the regulator begins to increase. This, in turn, causes the aneroid plate to move upward, and reduces the gap between the valve head and the seat, thus reducing the flow. When downstream demand ceases completely, the valve head is closed against the seat and the flow is stopped.

The regulator body is machined from 304 stainless steel. All other internal components exposed to the water, including the bellows, are also fabricated from stainless steel, except for the o-ring and the seat which are molded rubber. The end of the aneroid, which is not in contact with the water, employs a deposited nickel bellows as a dynamic seal for the adjustment feature of the aneroid.

The maximum water flowrate is 36.3 Kg/hr (80 lbs/hr) with inlet pressure ranging from 106.5 to 828 KPa (15.5 to 120 psia) and outlet pressure of 103 ± 3.5 Pa ($15 \pm 1/2$ psia). Minimum operating life is 20,000 hours and mini-

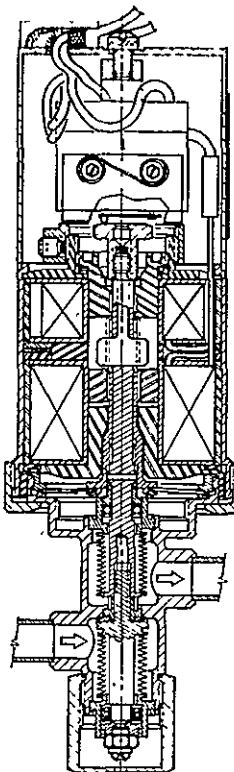


FIGURE 3-10 ISOLATION VALVE CROSS SECTION

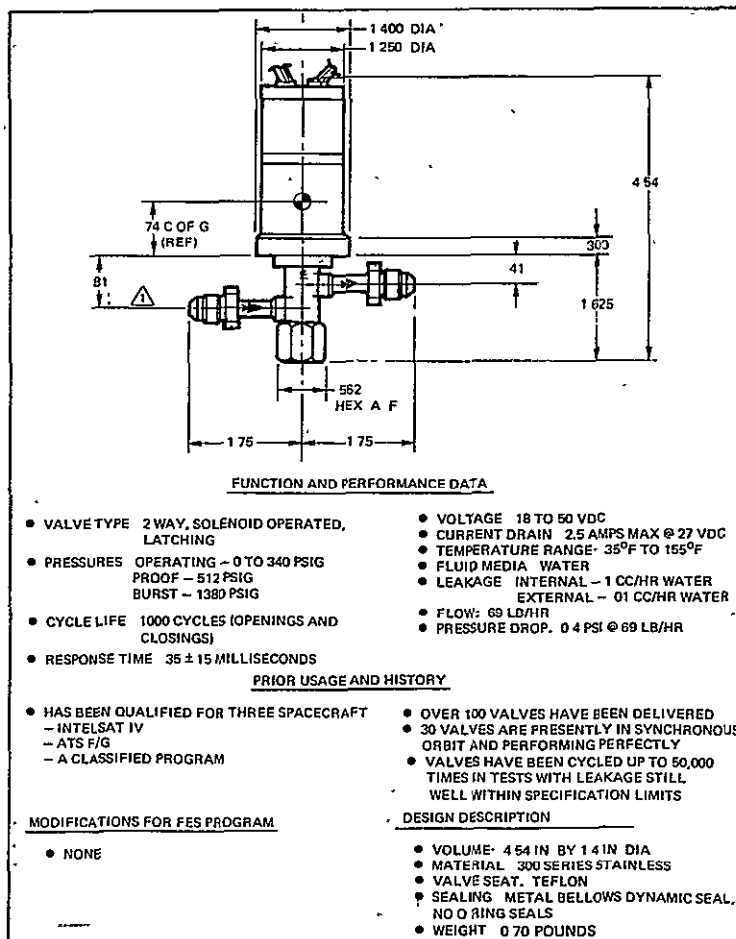
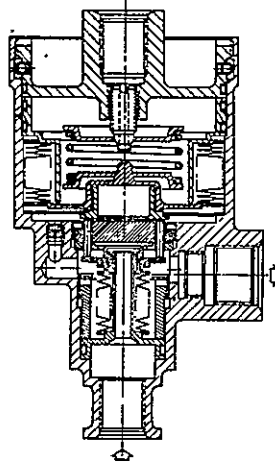


FIGURE 3-11 ISOLATION VALVE



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FIGURE 3-12 WATER PRESSURE REGULATOR

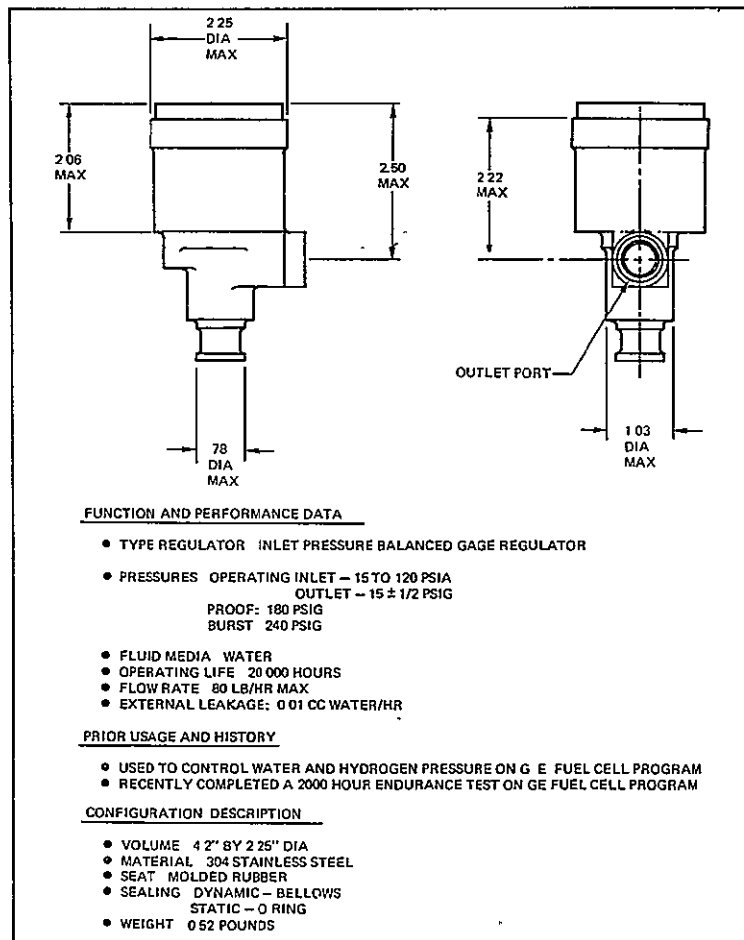


FIGURE 3-13 WATER PRESSURE REGULATOR

imum shelf life is 10 years. External leakage is no greater than 0.01 cc of water per hour at maximum inlet pressure. Internal leakage is no greater than 2.0 cc of water per minute at 110 KPa (16.0 psia) outlet pressure and at any inlet pressure up to 828 KPa (120 psia). A summary of the performance data, prior usage, and design description is given in Figure 3-13.

THERMISTOR PROBE ASSEMBLY

The thermistor probe is a two bead design with a probe which fits into a thermowell. The design permits removal of the probe assembly without loss of fluid from the system. This design is a modification of an existing thermistor probe which is currently being supplied for the Space Shuttle ECS System. The outline and installation configuration is shown in Figure 3-14. The probe metal parts are fabricated from 316 CRES and, when installed in the thermowell, is hermetically sealed. The thermowell is fabricated from 6061 aluminum. To improve the sensor time constant, the thermistor beads are exposed. However, as shown on Figure 3-14, the beads are recessed from the probe tip to prevent accidental damage during handling, calibration and installation.

For installation, the thermowell is filled with Dow Corning #340 Silicone Heat Sink Compound. When the probe assembly is threaded into the well, the silicone grease fills all voids thereby improving the heat transfer between the well and the thermistor beads. Using this method, results of 1.9 - 2.0 seconds were attained for 63% response to a step change in fluid temperature. The thermistor beads are held in place using Emerson & Cuming Eccobond 285 epoxy. This provides an adequate moisture seal but cannot be considered a hermetic seal in the true sense of a glass-to-metal seal or an all-metal seal.

Within the 4.76 mm (.187 in.) diameter sheath, the individual leads are insulated with thin Teflon sleeving. In the area where the sensor leads enter the cylindrical section of the housing, the leads are capped-off with an RTV compound. The remaining void under the connector back-face is filled with aluminum oxide powder to support the uninsulated leads and to eliminate movement resulting from sustained vibratory forces. The hermetic receptacle is fabricated from 321 stainless steel and is TIG welded to the sensor housing.

The thermistor bead chosen for this assembly is a 10K ISO-CURVE thermistor. It is a close tolerance, low drift device with exceptional response time and self-heating characteristics. The bead is actually an assembly of two beads whose R-T characteristics combine to meet the desired resistance-temperature curve.

SUPERSONIC NOZZLE

The supersonic nozzle must eject evaporated vapor overboard at Mach 3 or greater at mass flowrates of .453 Kg/hr (1.0 pph) or greater. The nozzle must be provided with electrical heaters to maintain the temperatures greater than 1.7°C (35°F) and it must withstand maximum temperatures of 1198.89°C (2190°F)

The supersonic nozzle design is shown in Figure 3-15; it is made from Haynes Alloy 188 (H-188). This material was selected over Columbium C-103

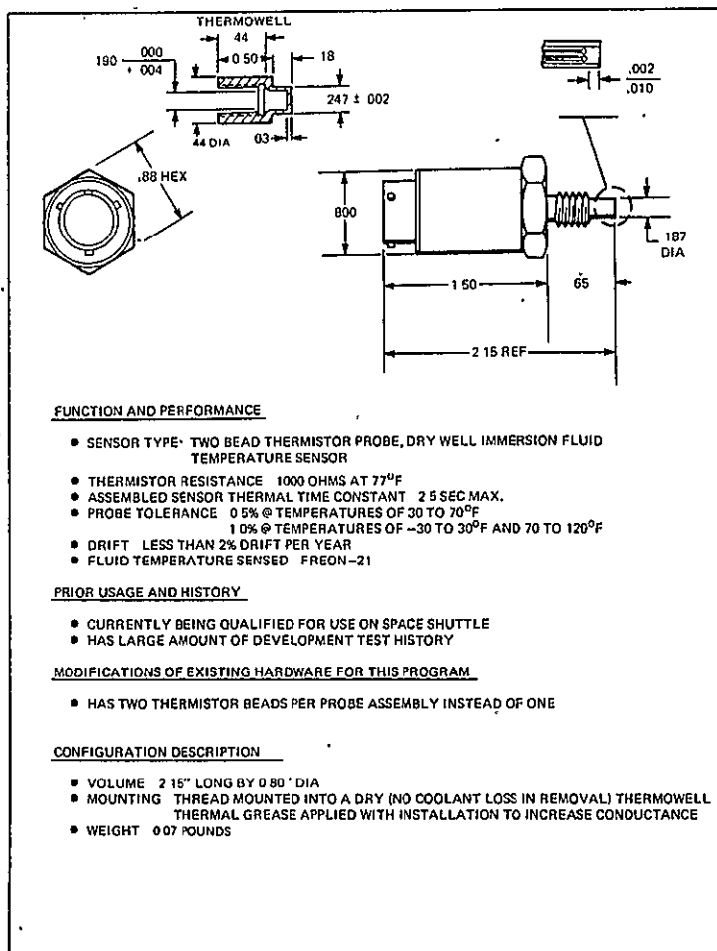


FIGURE 3-14 THERMISTOR PROBE ASSEMBLY

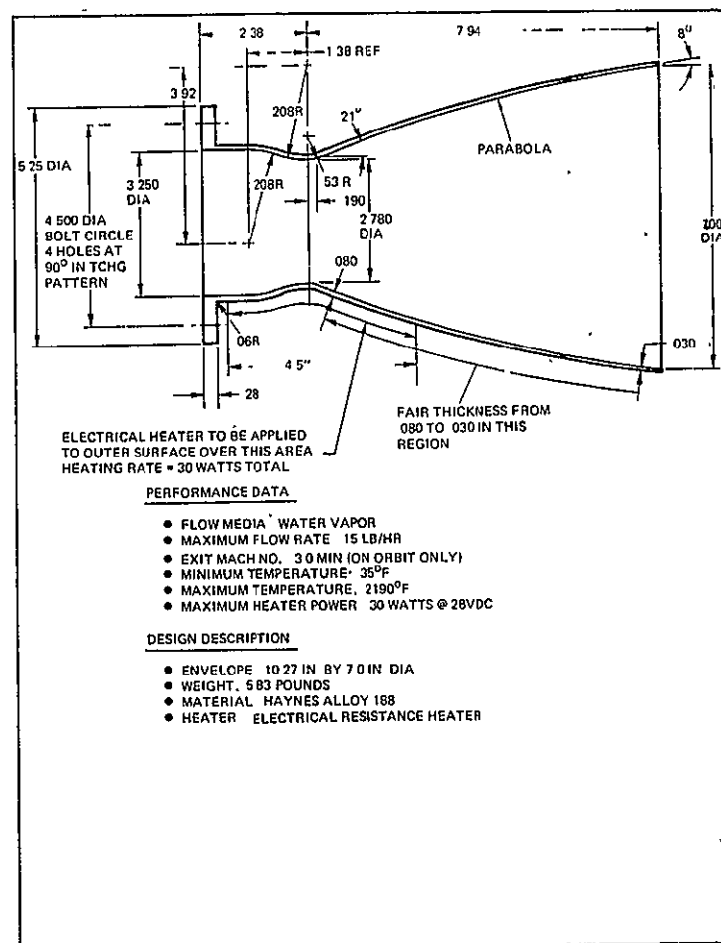


FIGURE 3-15 SUPERSONIC NOZZLE

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because; (1) H-188 has sufficient strength at the specified temperature of 1198.89°C (2190°F), which Vought regards as having adequate margin compared to expected temperatures, (2) no oxidation inhibiting coating is required, and (3) substantially lower cost.

ELECTRONIC CONTROLLER

The primary function of the controller is to monitor the temperature at the flash evaporator control point with a thermistor bead submersed in the heat transport fluid and to supply 28 volt pulses to the water metering valves at a rate required to achieve the required control temperature. Control is achieved by varying the pulse width of pulses to the valve at a rate proportional to the sensed temperature error. The pulse frequency is constant at 6 seconds between the start of each pulse. One controller performs this control function for both the high and low temperature portion of one Freon loop. Each temperature control circuit contains a redundant circuit. Four temperature control circuits exist for each of the two controllers. The high temperature unit controls to 15.6°C (60°F) and the low temperature unit controls to $2.8 \pm 1.1^\circ\text{C}$ ($37 \pm 2^\circ\text{F}$).

In addition to the control function, the controller deactivates the high and low temperature units based upon inlet temperature. The low temperature unit deactivates the low temperature control circuit when inlet temperature is less than 4.4°C (40°F) and activates it when greater than 5.0°C (41°F). The high temperature unit deactivates the control circuit when the inlet is less than 16°C (61°F) and activates it when greater than 16.7°C (62°F).

A third function of the controller is the monitoring of control fidelity and supply of a warning signal if faulty operation is detected. This function is performed by monitoring the outlet temperature and if out of control for a specified length of time; a warning signal is given. If later control is achieved and maintained for a specified time, the warning signal is turned off.

3.4 Design Analyses and Optimization

The design analyses used to optimize the RSEHS heat exchanger core, spray nozzle, and connecting fluid plumbing to assure the RSEHS will meet the performance requirements of Section 3.1 are presented in this Section.

HEAT EXCHANGER CORE OPTIMIZATION

The optimum heat exchanger core for the FES minimizes cost and total system weight including structural mass and weight equivalences for fluid pressure losses. The variables to be considered in selecting the optimum core include the fin geometry and material, the measured core performance relating pressure losses and convection coefficients, and the required heat transfer rates set by the spray nozzle and evaporator geometry. Figure 3-16 illustrates the flow of information used for selecting the best heat exchanger core for the RSEHS.

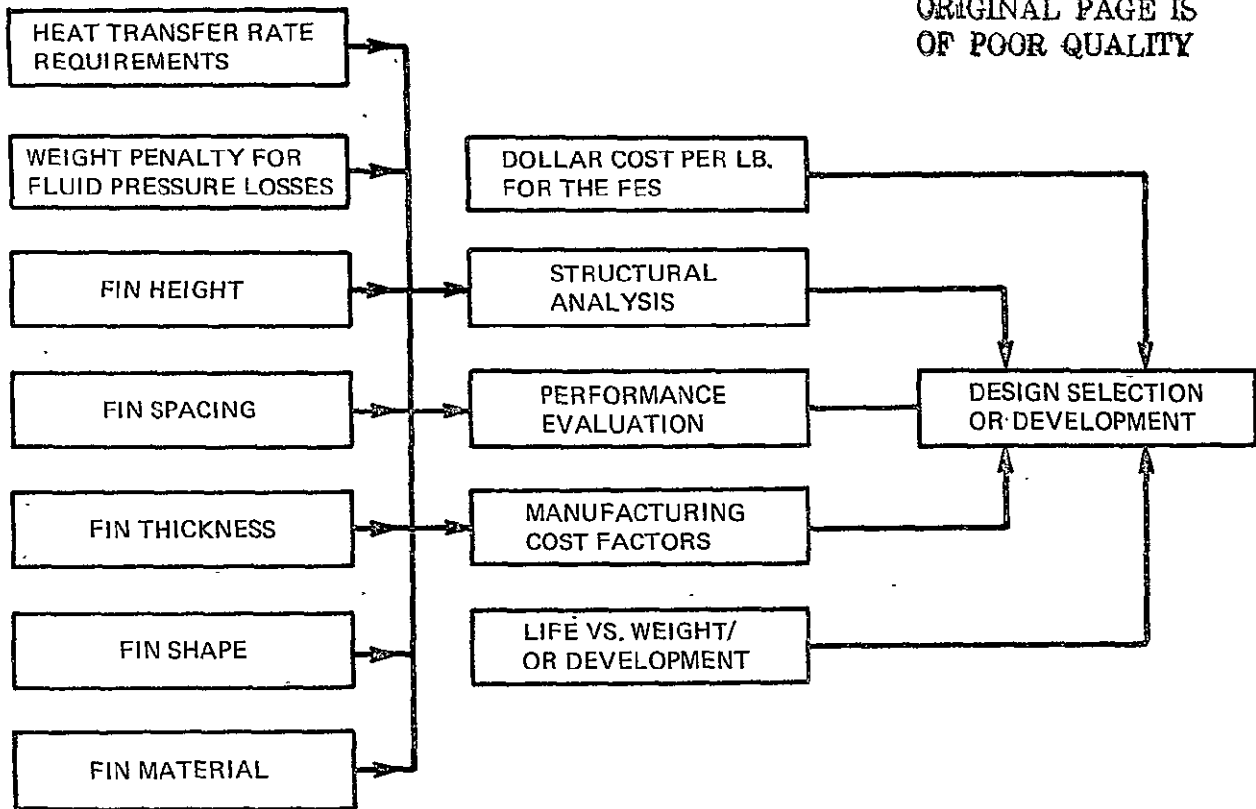


FIGURE 3-16 HEAT EXCHANGER CORE OPTIMIZATION APPROACH

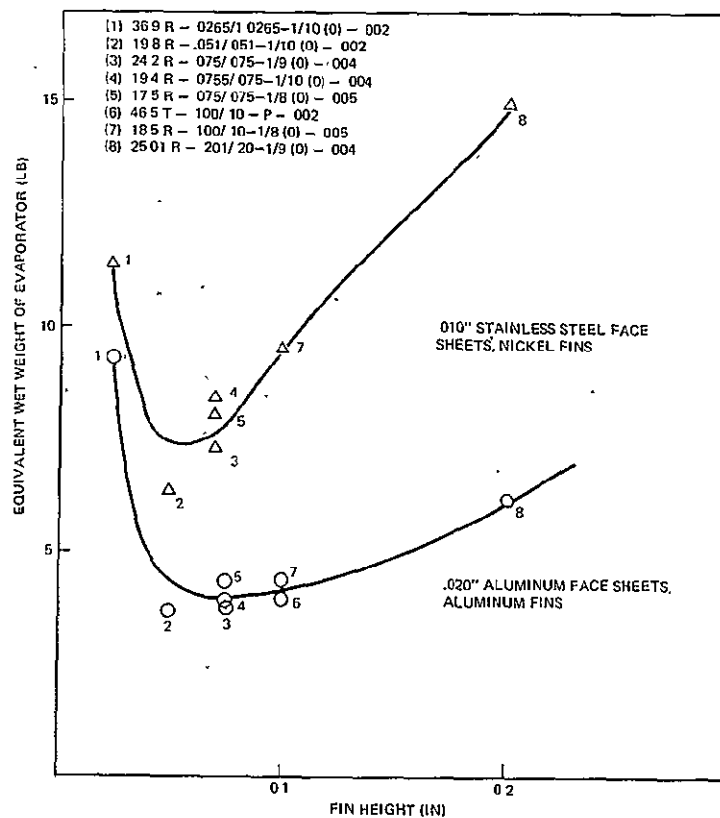


FIGURE 3-17 EQUIVALENT WEIGHT VS FIN HEIGHT

The requirements for the core of the FES differ from those of other liquid-to-liquid heat exchangers in that the local heat transfer rates are set by the spray nozzle and must be matched to the core performance. Since the evaporator walls contain only one layer of core, the flow area is somewhat restricted and the pressure losses become prohibitively large for cores with short fins. Thus, the weight penalty for additional pumping power more than off-sets the core weight savings made possible with these fins. As a result, heat exchanger cores with very short fins which typically are best suited for liquid systems are not optimum in this case.

Figure 3-17 gives the equivalent wet weight of the heat exchanger core and face sheets for several evaporator designs as a function of fin height for 8 different cores. The equivalent weight contains a weight penalty factor of 0.18 g/Pa (2.7 lb/psi) for pressure losses. The weight penalty is computed assuming a pump efficiency of 33% and an electrical power generation weight factor of 240 g/w (530 lb/kw). The figure shows that the equivalent wet weight of aluminum cores is minimized with fin heights near 1.9 mm (0.075 in.). The 2.54 mm (0.100 in.) high fins were selected because they provide additional structural stiffness at very little cost in equivalent core weight over shorter fins. Stainless steel/nickel cores were considered as an alternate to aluminum. However, the FES environment and operating pressure do not justify the additional core weight and volume required for this construction. Figure 3-17 shows that the core of a stainless steel top-off unit would weigh approximately four pounds more than the baseline aluminum core. The weight of the backcone and supporting structure would also increase so that the system would be considerably heavier if stainless steel/nickel cores were actually required.

Detailed design of the RSEHS heat exchangers is based on conventional heat exchanger design techniques on the Freon side. Heat transfer coefficients and pressure drops are based on j-factor and f-factor data for the heat exchanger core material, which is a 2.54 mm (0.1 in.) high, 3.175 mm (1/8 in.) long lance fin with 0.73 fins/mm (18.5 fins per in.). The maximum evaporation heat transfer rate is dependent on surface temperature and core Freon heat transfer coefficient. Evaporation rates are based on data from element tests conducted in Reference 1 which are shown in Figure 3-18. A value of 1.14 watts/cm²-°K (2000 BTU/hr-ft²-°F) is used for the evaporation heat transfer coefficient to provide margin. The critical value at which icing or flooding occurs is above 2.27 w/cm²-°K (4000 BTU/hr-ft²-°F). The local region heat transfer rate

$$\frac{dq}{dA_w} = U_o(T_w - T_s) \quad (1)$$

$$U_o = \frac{1}{\frac{1}{nh_e \left(\frac{A_o}{A_w}\right)} + \frac{x}{k \left(\frac{A_w}{A_w}\right)} + \frac{1}{h_e \left(\frac{A_w}{A_w}\right)}}$$

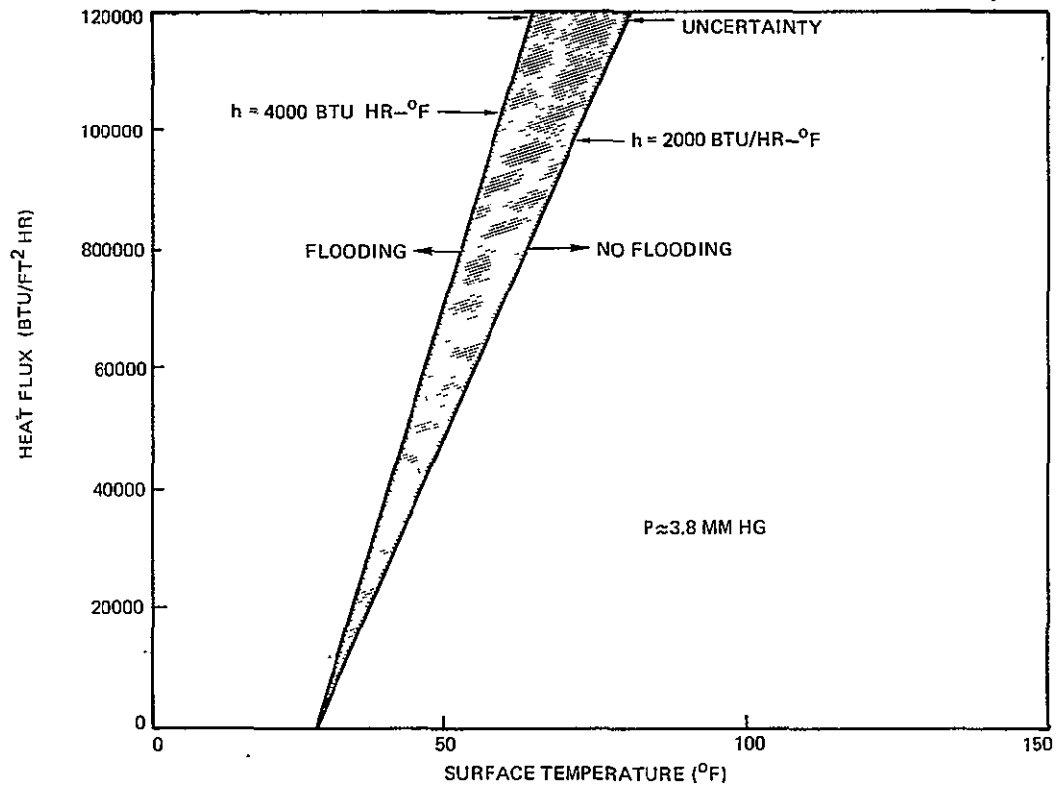


FIGURE 3-18 MAXIMUM FES EVAPORATOR HEAT FLUX

where:

q	=	heat transfer rate
A	=	Area
U_o	=	overall heat transfer coefficient
x	=	wall thickness
T_w	=	heat exchanger wall temperature
T_s	=	spray chamber saturation temperature
η	=	heat exchanger fin effectiveness
h_c	=	inside heat transfer coefficient, based on fin j-factor data
h_e	=	evaporation heat transfer rate
A_o	=	total heat transfer area inside the heat exchanger core
A_w	=	area of heat exchanger surface

This establishes the maximum water spray density which can be applied to the load region dA_w , since

$$\frac{dq}{dA_w} = \frac{d\dot{w}}{dA} L \quad (2)$$

so, combining equations (1) and (2)

$$\frac{d\dot{w}}{dA} = \frac{U_o(T_w - T_s)}{L} \quad (3)$$

where:

$\dot{w}$	=	water spray rate
L	=	latent heat of vaporization of water

It is then necessary to know the spray distribution of the nozzle. The spray forms a solid angle from a point, so the density of the spray decreases as the square of the distance from the nozzle. With the distribution data the required distance from the nozzle to the heat exchanger core can be established. This also establishes the diameter of the cylindrical section of the heat exchanger. An iterative process is used to optimize the size because changes to the diameter of the heat exchanger cylinder change flowrate and thus heat transfer coefficient on the Freon side.

SPRAY NOZZLE OPTIMIZATION

Spray nozzles have been developed specifically for the RSEHS which optimize the distribution of the evaporant on the heat exchanger core, and which operate reliably and efficiently in a vacuum environment. These nozzles result in minimum core dimensions and system weight, and produce consistent uniform spray patterns at the supply pressures and operating environment of the Space Shuttle.

To achieve an efficient and reliable FES design the spray nozzles must be constructed such that their operating characteristics remain within very narrow and exactly specified limits. Small variations in the nozzle performance have large impacts on the system size and characteristics. If the

spray is concentrated in a narrow area or is non-uniform and unsymmetrical, the evaporator dimensions must be made unnecessarily large to prevent localized ice formation in areas where the cooling rates are too high. Ice can form on the nozzle face, which can chip off at a substantial rate and accumulate in the evaporator cavity. Ice which forms inside the nozzle during non-flow periods at partial loads temporarily changes the flow resistance through the nozzle and interacts with the electronic controller so as to cause unacceptable perturbations in the transport fluid outlet temperature.

Based on the experience gained in testing the standard nozzles obtained from 8 manufacturers and from further analyses and element tests conducted to determine the effects of specific design variables on the nozzle performance in a vacuum, nozzles were constructed and tested which do satisfy all the desired requirements. The special nozzles have operating characteristics ideally suited for the RSEHS. The ideal nozzles distribute the spray on the heat exchanger core such that maximum cooling occurs at each location consistent with the evaporation potential provided by the transport fluid temperature. Thus, the maximum possible quantity of heat is withdrawn from each elemental area and the total required heat exchanger core area is minimized. The maximum heat transfer rate is given by the following equation

$$dq_{\max} = U_o(T-T_s) dA \quad (4)$$

where: U_o = the overall heat transfer coefficient including the convection coefficient within the core and the evaporation rate coefficient
 T = transport fluid temperature
 T_s = evaporant saturation temperature
 dA = incremental area of heat exchanger core

The energy equation for the transport fluid is

$$dq_{\max} = -\dot{m} C_p dT \quad (5)$$

Solving Equations (4) and (5) simultaneously yields

$$T-T_s = (T_{in} - T_s) e^{-\frac{U_o A}{\dot{m} C_p}} \quad (6)$$

where: T_{in} = inlet temperature of transport fluid
 A = area measured from transport fluid inlet

Figure 3-19 compares the experimental transport fluid temperature obtained with the specially designed nozzles to those possible with maximum cooling as given by Equation (6). The actual fluid temperature profiles are very similar to the ideal ones thus demonstrating that the nozzle spray pattern is nearly optimum. The evaporator dimensions are set such that the actual cooling rates are slightly less than the maximum possible values to allow a safe operating margin for off design requirements. Thus the actual fluid

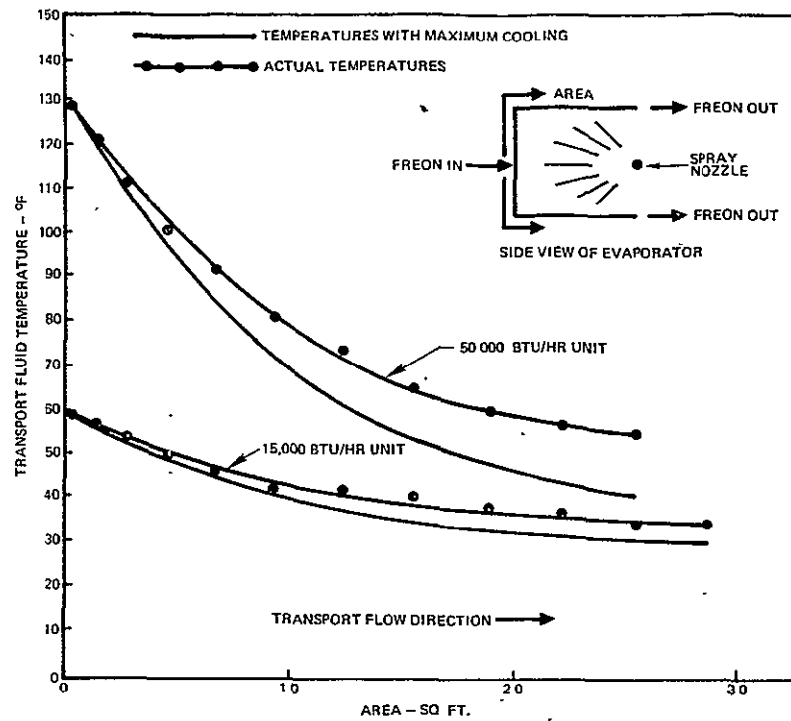


FIGURE 3-19 COMPARISON OF ACTUAL AND IDEAL NOZZLE SPRAY DISTRIBUTION

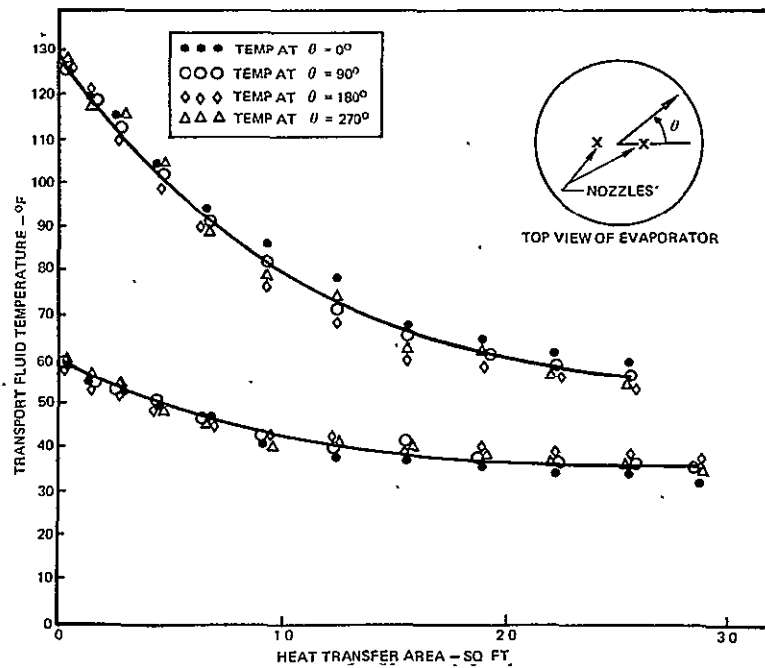


FIGURE 3-20 SPRAY PATTERN UNIFORMITY IN REDUNDANT NOZZLE CONFIGURATION

temperatures are maintained above the values for which icing would occur. The temperature of the 14.6 kw (50,000 BTU/hr) unit appear to be unnecessarily conservative, but the larger operating margin near the exit of this unit actually is an accumulative result of all of the local upstream cooling rates being maintained only slightly below the maximum value.

An ideal nozzle would produce a spray pattern which is uniform and symmetrical with respect to the nozzle centerline. This would minimize the possibility of forming ice in localized areas where the spray density is concentrated. Uniform spray patterns are difficult to produce at the low supply pressures required in the Space Shuttle. Also the nozzles must be offset from the evaporator centerline to permit the use of redundant valves and nozzles. This introduces additional local variations in the spray density which must be accounted for by increasing the heat transfer area so as to reduce the average cooling rate. Tests were conducted to demonstrate that the nozzle spray patterns are sufficiently uniform to permit the use of redundant nozzles with the proposed evaporator core dimensions. The tests were completely successful. Steady-state and transient conditions were tested with the nozzle positioned off centerline and with the transport fluid outlet temperature being controlled at a set point below that required for the Space Shuttle. No operating difficulties were experienced. Figure 3-20 shows that the local fluid temperatures measured at corresponding positions on the evaporator floor and sidewalls do not vary significantly from the average value. This illustrates that the spray pattern is very uniform and that redundant nozzles can be accommodated with this proposed design.

PRESSURE DROP ANALYSES

Detailed pressure drop analyses were performed on the proposed FE assembly installation to determine the capability to meet the FES performance requirements of 31 KPa (4.5 psi) at 1089 Kg/hr (2400 lb/hr) Freon flow. The FE assembly was broken up into 28 segments, as shown in Figure 3-21, to perform the analyses. A summary of the results, presented in Table 3-3, indicates a total assembly pressure drop of less than 20.68 KPa (3 psi) at 1089 Kg/hr (2400 lb/hr) Freon flow which is less than the 31 KPa (4.5 psi) allowed.

TABLE 3-3

LOCATION	DESCRIPTION	D _H	L	K	M pph	ΔP psi
1-2	FLEX CONN	3/4"	2"		2400	.0045
2-3	LINE, TURN, & THERM.	3/4"	10"	66	2400	.0863
3-4	EVAP INLET				2400	.2500
4-5	EVAP BOTTOM	0.064"			2400	.4500
5	BOTTOM TURN	.2"		2	2400	.0011
5-6	CORE SIDE	0.064"	3"		2400	.0166
6	MANIFOLD ENTRY	.2"		1.0	400	.0005
6-7	MANIFOLD	1/2"	4'		200	.0013
7-8	INLET TO SPOKE	3/8"		1.25	400	.0753
7-8	SPOKE TUBE	3/8"	5.9'		400	.0329
8	SPOKE OUTLET	3/8"		1.25	400	.0753
8-9	UPPER MANIFOLD	3/4"	4.3'		1200	.0037
9	COLLECTOR ENTRY TUBE	1/2"		1.44	675	.0625
10-11	TUBE	1/2"	2 1/2"		675	.0068
11	MANIFOLD ENTRY	1/2"		1.0	675	.0434
11-12	CORE MANIFOLD	1/2"	3.5"		387	.0028
12	CORE ENTRY	0.064"		1.5	675	.0028
12-13	CORE	0.064"	3.2'		675	.0253
13	MANIFOLD ENTRY	0.2"		1.0	675	.0005
13-14	MANIFOLD	1/2"	7.4"		337	.0060
14	ENTRY	1/2"		1.44	675	.0625
14-15	TUBE	1/2"	3.6"		675	.0098
15	ENTRY TO CONNECT TUBE	3/4"		5	675	.0217
15-16	LINE, TURN, & THERM	3/4"	7.2"		2400	.0204
16-17	EVAP INLET				2400	.2500
17-18	EVAP BOTTOM	0.064"			2400	.4500
18	BOTTOM TUBE	.2"		2	2400	.0011
18-19	EVAP SIDE	0.064"	5.6'		2400	.0333
19	MAN. ENTRY	.2"		1.0	2400	.0005
19-20	MANIFOLD	1/2"	4'		200	.0013
20	SPOKE INLET	3/8"		1.25	400	.0753
20-21	SPOKE TUBE	3/8"	5.9"		400	.0329
21	SPOKE OUTLET	3/8"		1.25	400	.0753
21-22	UPPER MANIFOLD	3/4"	4.3"		1200	.0037
22	COLLECTOR ENTRY	1/2"		1.44	1000	.1370
22-23	TUBE	1/2"	2"		1000	.0108
23	MANIFOLD ENTRY	1/2"		1.0	1000	.0953
23-24	CORE MANIFOLD	1/2"	2.7'		500	.0043
24	CORE ENTRY	0.064"		1.5	1000	.0103
24-25	CORE	0.064"	3.2"		1000	.0613
25	CORE EXIT	0.2"		1.0	1000	.0014
25-26	MANIFOLD	1/2"	5.9'		500	.0092
26	ENTRY TO CONNECTING TUBE	1/2"		1.44	1000	.1370
26-27	CONNECT TUBE	1/2"	10"		1000	.0542
27	ENTRY	3/4"		5	1000	.0076
27-28	LINE, TURN, & THERM	3/4"	10"	66	2400	.0863
28-29	FLEX CONN	3/4"	2"		2400	.0045
TOTAL ΔP						2.8046

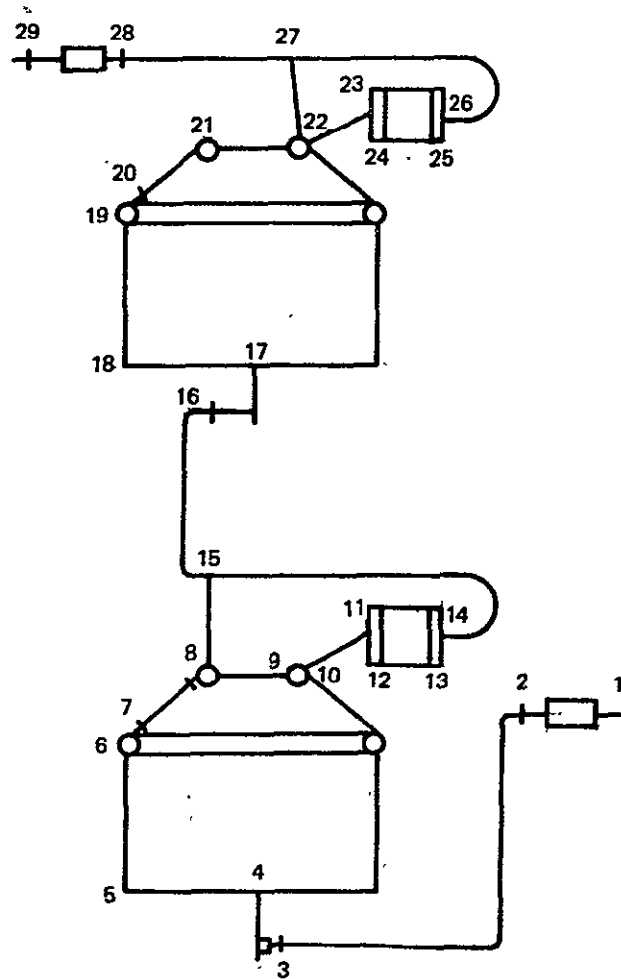


FIGURE 3-21 PRESSURE DROP FLOW SCHEMATIC

TOP-OFF SYSTEM NOZZLE DESIGN

The exhaust nozzle of the top-off evaporator is designed to produce a minimum exit Mach number greater than the required value of 3.0, and has overall dimensions compatible with the envelope specified. The nozzle exit diameter is made as large as possible within the constraints of the envelope to minimize the weight of the exhaust duct. The nozzle is designed to exceed the specified minimum value for exit Mach number at nominal conditions to insure that the required exit velocity is maintained at low evaporant mass flowrates. As is shown in the following analysis, this operating margin can be provided with very little increase in exhaust duct weight.

Figure 3-22 gives the critical dimensions of the nozzle designed for the top-off evaporator system. The wall profile was determined from a well established procedure developed by G.V.R. Rao. The converging section accelerates the flow from the exhaust duct, which has been maintained at a relatively low velocity to minimize pressure losses, to sonic conditions at the nozzle throat. It also diminishes the boundary layer which has developed in the duct so that a high nozzle efficiency is achieved. Turbulent boundary layers undergo reverse transition within the converging section when values of the parameter

$$K = \frac{v}{u^2} \frac{\partial u}{\partial x}$$

exceed about 2×10^{-6} . For the proposed nozzle K exceeds the critical value by an order of magnitude at the worst operating conditions so that the turbulent conditions in the duct will not influence the flow within the nozzle. The throat Reynolds number is relatively low because of the low pressures required by the evaporators. Therefore the nozzle discharge coefficient is not as high as can be expected in most converging diverging nozzles. Data in the literature indicate that the discharge coefficient for the maximum flowrates in the top-off unit is

$$C_D = 0.94$$

This means that the pressure upstream of the nozzle will be approximately 6% higher than is predicted from isentropic relationships. This higher upstream pressure must be accounted for when designing the duct. The nozzle efficiency will also be lower than is typical of supersonic nozzles, especially at low load conditions when the evaporant flowrate is of the order of one pound per hour. The Reynolds number for this case is approximately 1000 so that the nozzle efficiency could drop below 90%. To provide for high exhaust velocities at low load conditions the nozzle is designed assuming a worst case efficiency of 85%. Therefore at nominal loads the exit Mach number will exceed the minimum required value. The maximum Mach number is

$$M = \frac{3.0}{0.85} = 3.25$$

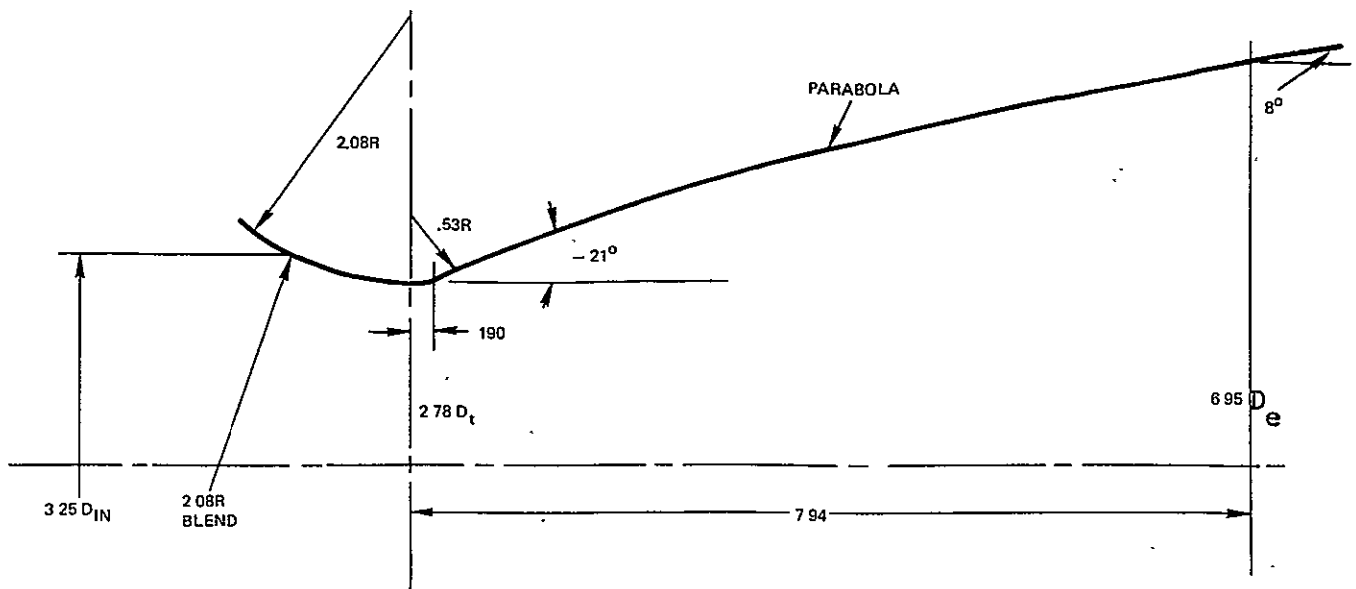


FIGURE 3-22 TOP OFF NOZZLE CRITICAL DIMENSIONS

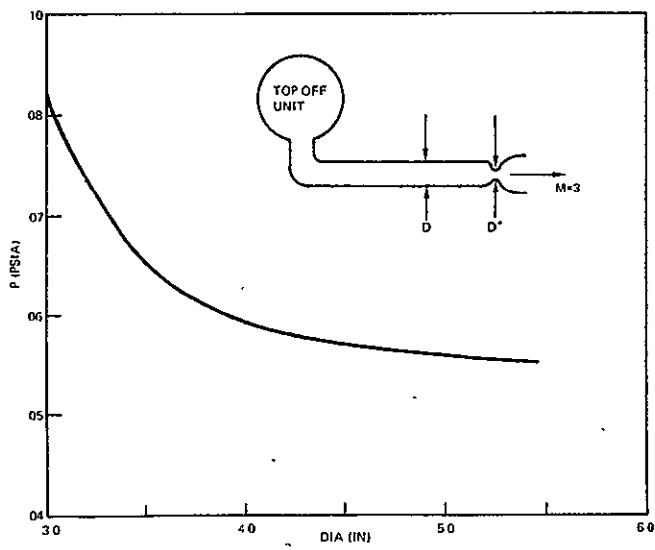


FIGURE 3-23 EVAPORATOR
CHAMBER PRESSURE VS DUCT DIAMETER

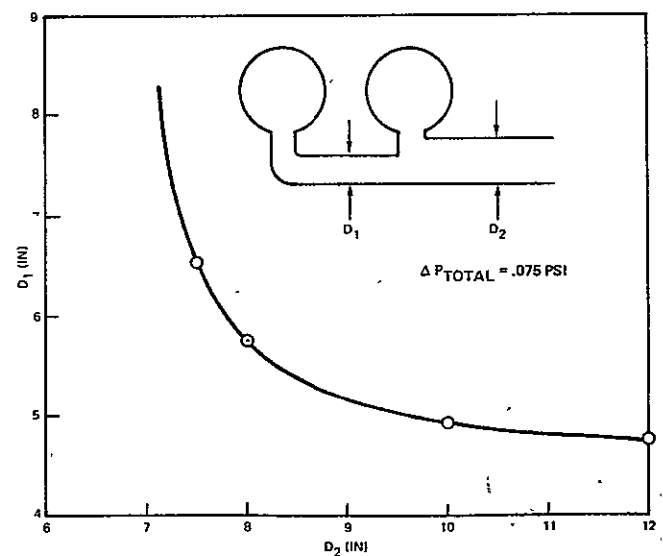


FIGURE 3-24 DIAMETER OF THE DUCT

Thus the throat diameter can be sized from isentropic relationships and the known diameter at the nozzle exit. For a 177.8 mm (7") O.D. exit diameter and $M_E = 3.25$

$$D^* = \frac{D}{A/A^*} = \frac{6.95}{6.28} = 2.78 \text{ in.} = 70.61 \text{ mm}$$

TOP-OFF SYSTEM DUCT DESIGN

Since the Mach number at the exit of the exhaust duct is expected to be less than 0.5, the static pressure is only slightly less than the total pressure. The total pressure at the exhaust duct exit can be determined from isentropic relations for choked flow in the nozzle. In calculating the upstream pressure the maximum flow rate through each nozzle is assumed to be 17 lb/hr. The stagnation temperature is established by the conditions within the evaporator and the amount of heating in the exhaust duct. However, it will always be maintained at a minimum of 40°F to prevent condensation in the duct. Thus the stagnation pressure is

$$P_0 = \frac{\dot{w}}{A^*} \frac{RT}{K} \left(\frac{K+1}{2} \right)^{\frac{K+1}{K-1}} = 0.042 \text{ psi} = 0.29 \text{ KPa}$$

As discussed above, the actual upstream pressure must be greater than the ideal pressure to account for non-isentropic flow in the nozzle. Thus the actual upstream total pressure is taken to be

$$P_0 = (1.06)(0.042) = 0.045 \text{ psi} = 0.31 \text{ KPa}$$

The static pressure will be slightly less, but for conservatism will be assumed to be the same as the stagnation pressure. The pressure inside the evaporator chamber is determined by adding the pressure loss in the duct to the pressure upstream of the nozzle.

$$P_{(\text{CHAMBER})} = .045 + \Delta P (\text{DUCT}) \text{ psi} \quad (7)$$

Figure 3-23 gives the evaporator chamber pressure computed from Eq. (7) as a function of duct diameter. The figure shows that the diameter required for a chamber pressure of 0.52 KPa (0.075 psi) is 80.04 mm (3.15").

The pressure drop in the duct was computed by numerically integrating the incremental losses in small sections. For straight sections the head loss was computed with the local density, velocity, and friction factor

$$\Delta P = \frac{1}{2D} \int_0^{\Delta L} \rho V^2 f dx \quad (8)$$

For curved sections, loss coefficients taken from the literature were employed.

$$\Delta P = \frac{K}{2} \rho V^2. \quad (9)$$

In equations (8) and (9) the local velocity and density were computed from ideal gas relationships and the continuity equation assuming isothermal flow in the duct. The local pressure was computed by summing the head losses between the given section and the nozzle entrance. There is some uncertainty in the analysis because the exact values for the loss coefficients in Eq. (9) are unknown. For conservatism the bend loss coefficients were assumed to be slightly larger than the values estimated from the reference data, and the duct was designed for a flow rate that is 10% higher than required.

RE-ENTRY DUCT DESIGN

The duct connecting the two re-entry units carries only half of the total flow. Therefore it is advantageous to make the diameter of the connecting duct smaller than that of the duct which must carry the total flow. The diameters of the two ducts must be related such that the head loss across both ducts is less than a prescribed value. Therefore as the diameter of the larger duct is decreased, the diameter of the connecting duct must increase to maintain a constant upstream evaporator chamber pressure. Figure 3-24 gives the required diameter of the connecting duct as a function of the downstream duct diameter. The connecting duct diameter becomes very large when the pressure loss in the downstream duct approaches the maximum permissible value. Also, the downstream duct must be very large if the connecting duct is small. Therefore the combined weight of the ducts can be minimized by selecting an optimum combination of duct diameters. Figure 3-25 shows that the optimum diameter of the larger duct is approximately 215.9 mm (8.50"). The corresponding diameter of the connecting duct determined from Figure 3-24 is 139.7 mm (5.50").

The pressure losses in the re-entry system ducts were calculated from Equations (8) and (9) using the procedure described above. The pressure at the exit of the large duct is calculated assuming choked flow at the exit plane. Thus the duct diameter depends on the velocity of the exhaust gases at this position. The value of the velocity at the exit depends slightly on the characteristics of the flow in the duct. For example, for isentropic flow

$$V^* = 1260 \text{ ft/sec} = 384 \text{ m/sec}$$

For Fanno flow,

$$V^* = 1270 \text{ ft/sec} = 387.1 \text{ m/sec}$$

For isothermal flow,

$$V^* = 1180 \text{ ft/sec} = 359.7 \text{ m/sec}$$

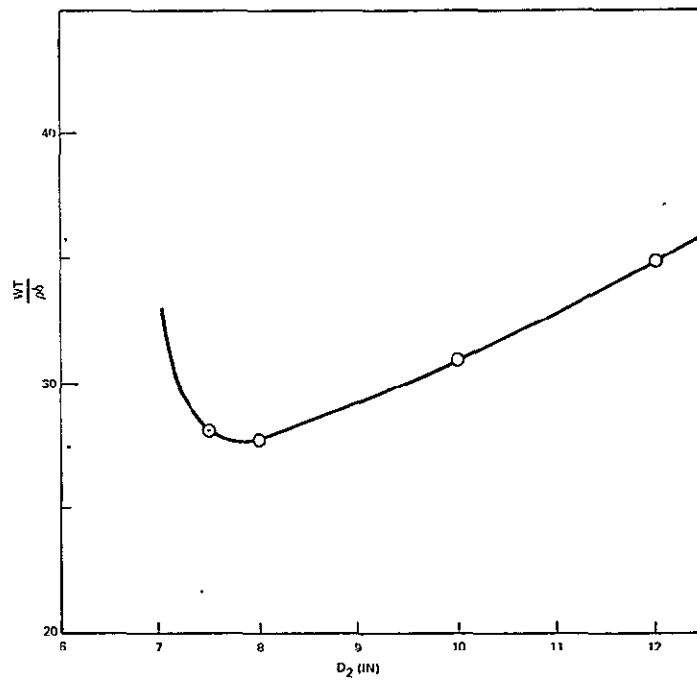


FIGURE 3-25 DOWN STREAM DUCT OPTIMIZATION

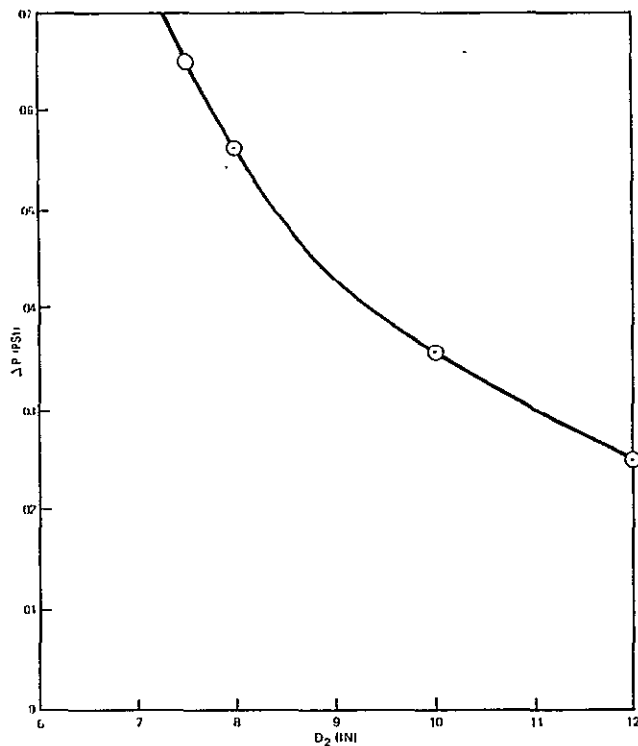


FIGURE 3-26 RE-ENTRY SYSTEM DUCT
PRESSURE LOSS

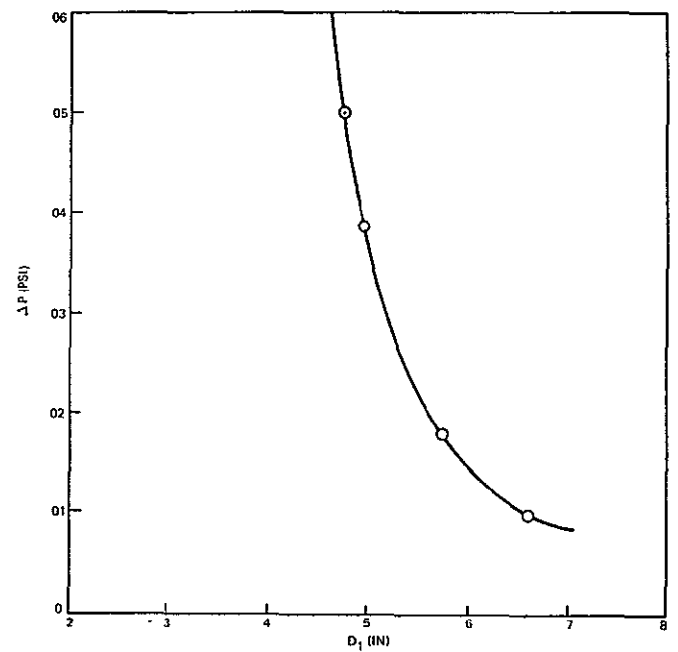


FIGURE 3-27 RE-ENTRY SYSTEM DUCT
PRESSURE LOSS

The flow in the actual case is not easily classified. Therefore, for conservatism in the design it is assumed that the exit velocity is only 335.28 m/sec (1100 ft/sec). The density at the exit computed from the continuity equation is

$$\rho^* = \frac{\dot{m}}{AV^*} = \frac{0.354 \cdot 10^{-4}}{D^2} \text{ lb}_m/\text{ft}^3 \quad (10)$$

The exit pressure is computed from the ideal gas equation of state

$$p^* = \rho^* R T^* = \frac{.0106}{D^2} \text{ psi} \quad (11)$$

The Reynolds number is

$$Re = \frac{4\dot{m}}{\pi\mu D} = \frac{6360}{D} \quad (12)$$

The conditions at the exit plane were computed for given duct diameters by solving Equations (10) to (12). The conditions upstream of the unit were then determined by integrating the pressure losses in small sections using the procedure developed above. When the pressure had been determined at the exit of the connecting duct by integrating the head losses over the entire length of the large duct, the diameter of the connecting duct was computed to give the required operating pressure in the inboard evaporator. The pressure losses computed for the ducts of the re-entry system are given in Figures 3-26 and 3-27.

4.0 RSEHS FABRICATION AND TESTING

4.1 RSEHS Fabrication

At the completion of RSEHS design, fabrication of one set of RSEHS heat exchanger cores and the metering valves was initiated. At this time, the FES for the Space Shuttle was awarded to another contractor and NASA issued a "stop work" order on further fabrication. The fabrication status at the time of the stop work order is reviewed in the following paragraphs.

One set of RSEHS heat exchanger cores were fabricated to demonstrate the capability of the RSEHS design approach to meet the performance requirements. The units were fabricated using salt brazing with aluminum 3003 core and face sheet material rather than the vacuum-fluxless brazing with 6000 series aluminum core and face sheet in order to obtain the heat exchanger core as soon as possible in the program in order to test the design.

The Freon distribution ring, collector ring, and 8 transfer tubes were welded to each core to provide a complete heat exchanger and fluid flow unit for testing. Additionally, four segments (half) of the low temperature unit backcone were fabricated and welded to the unit. The resulting high temperature and low temperature units are shown as fabricated in Figures 4-1 and 4-2 respectively.

At the time of work stoppage, the units shown had completed pressure drop checkout testing. The units achieved less pressure drop than calculated in Section 3.4.

All the evaporant metering valves were in fabrication and were subsequently completed at the time of the stop work order. The low and high temperature unit metering valves are shown in Figures 4-3 and 4-4. The low temperature unit spray nozzles which are a part of the valves were fabricated and integrated with the valve, but the resulting assembly was not flow checked. Fabrication of the high temperature unit nozzle was not initiated.

Fabrication was not initiated on any of the supersonic nozzles, exhaust duct elbow heat exchangers, back cones, valve/nozzle mounting plates, connecting shells, electronic controllers, honeycomb mounting plates, solenoid isolation valves, pressure regulators, or temperature sensors, for the second set of heat exchanger cores. Fabrication of an exhaust duct system in support of flash evaporator testing at NASA/JSC was accomplished under another contract. A description of the exhaust duct system design and fabrication is presented in Reference 2.

4.2 RSEHS Testing

4.2.1 Test Objectives

Nine months after the stop work order on the fabrication of the RSEHS system, NASA directed Vought to perform checkout testing of the

4-2



FIGURE 4-1 HIGH TEMPERATURE UNIT EVAPORATOR

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FIGURE 4-2 LOW TEMPERATURE UNIT EVAPORATOR



FIGURE 4-3 HIGH TEMPERATURE UNIT METERING VALVE

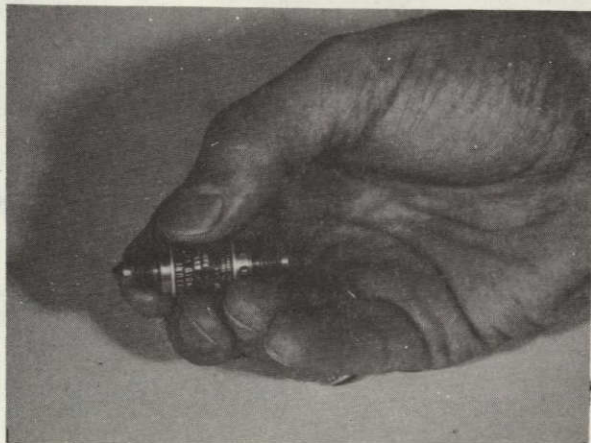


FIGURE 4-4 LOW TEMPERATURE UNIT METERING VALVE

RSEHS low temperature unit heat exchanger core to verify the heat rejection and operation of the unit as designed, and to demonstrate the unit capabilities for meeting high temperature unit heat rejection rates with a simple nozzle change.

The objectives of the testing were: a) demonstrate heat rejection performance at the design flowrate and temperature conditions with high and low temperature unit nozzles; b) demonstrate operation without frost formation on the heat transfer surfaces; c) demonstrate operation at partial load with a simulated controller.

4.2.2 Test Article and Instrumentation

The test article was the RSEHS low temperature unit shown in Figure 4-2, modified to accommodate a Prototype I flash evaporator metering valve, and RSEHS modified nozzles. As seen in the Figure, only partial backcone and no exhaust duct entry section were present on the unit. A decade resistance box was installed to simulate the controller to operate the water valve for partial load operation.

The test flow system is shown in Figure 4-5 along with instrumentation flow ranges and temperatures. In addition, 16 mm movies of flash evaporator operation at regular speed and with slow motion (32 fps) were taken.

4.2.3 Test Results

The results of the RSEHS testing are shown in Table 4-1. Five different flow conditions were run: three test points run with the low temperature unit nozzle and one test point run with the high temperature nozzle. The evaporator performed with high evaporation efficiencies considering that there was only half the backcone and no exhaust duct elbow.

The high temperature nozzle runs were made at maximum, 50% and 25% heat load conditions. The evaporator performed as expected with no ice formation on the core, and with evaporation efficiencies from 91.6 to 96.4%.

The low temperature nozzle runs were made at maximum heat load and at 50% heat load conditions. Frost formed on the core at the maximum heat load condition. The nozzle was found to be dirty. After cleaning, the test point was repeated with no frost formation on the core. The unit obtained an outlet temperature range of $3^{\circ} \pm 0.6^{\circ}\text{C}$ ($37^{\circ} \pm 1^{\circ}\text{F}$) during operation at both test points. This was better than the design requirement of $3^{\circ} \pm 1.1^{\circ}\text{C}$ ($37^{\circ} \pm 2^{\circ}\text{F}$). High evaporation efficiencies of 92.5 to 95.5% were obtained.

The RSEHS evaporator core met the test objectives by demonstrating the design performance, operation with no frost formation, and operation at partial load within the desired outlet temperature range of $3^{\circ} \pm 1.1^{\circ}\text{C}$ ($37^{\circ} \pm 2^{\circ}\text{F}$).

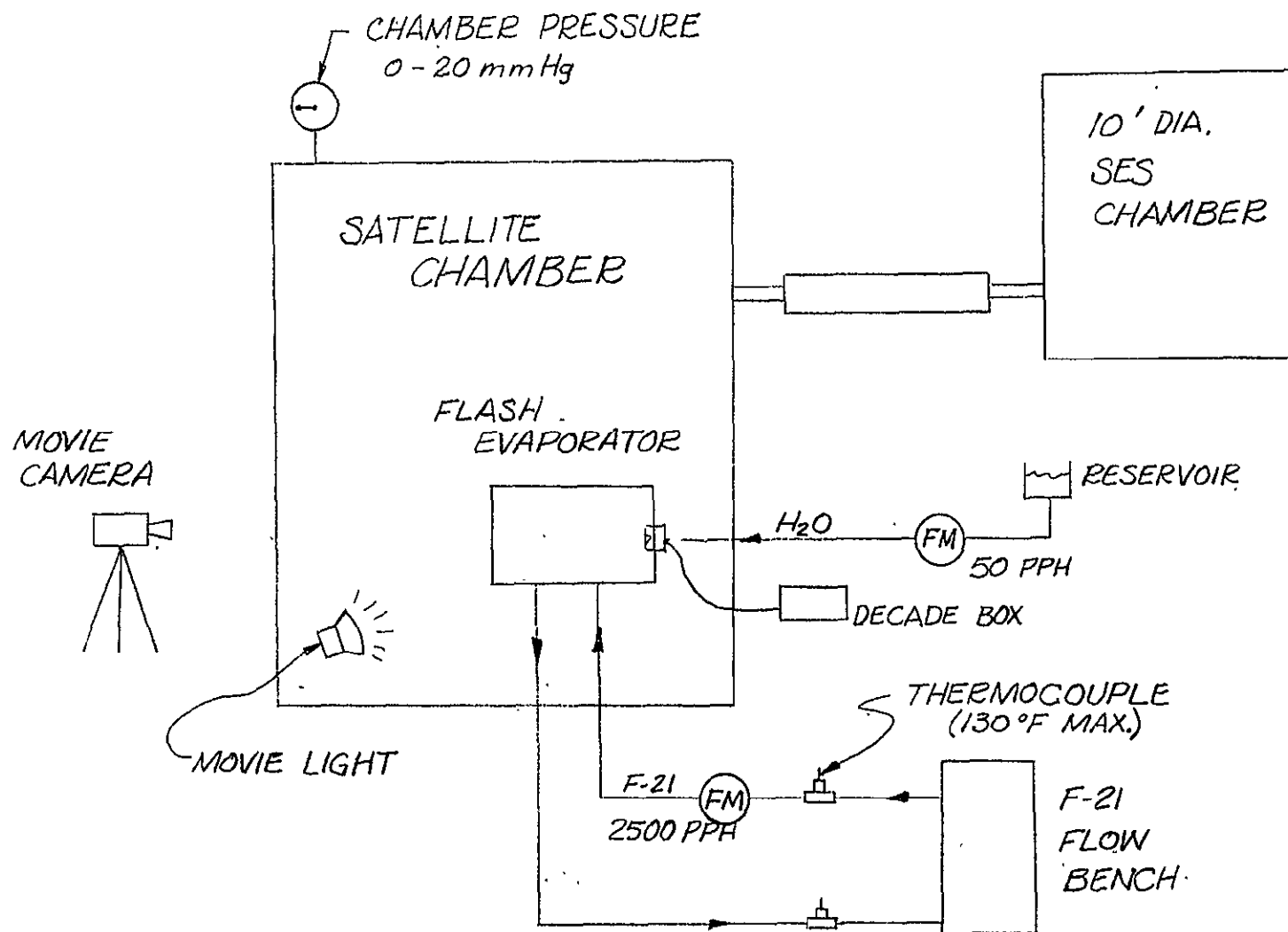


FIGURE 4-5 RSEHS TEST SET UP

TABLE 4-1 SPRAY TEST RESULTS OF RSEHS

m_{H_2O} (PPH)	m_{F21} (PPH)	T_{in} (°F)	T_{out} (°F)	Q_{ABS} (BTU/HR)	λ_{fg} (BTU/LB)	COMMENTS
14	2750	61	40	14207	1015	Ice on Manifold
15	2750	61	38	15560	1036	Ice on Manifold
16	2750	61	36.5	16574	1037	Ice on Manifold
14	2500	61	38	14145	1010	Ice on Manifold
15	2500	61	36.5	15375	1025	Ice on Manifold
16	2500	61	33.5	16913	1057	Ice on Manifold and Bottom at 6.00
14	2250	60	35	13840	988	Ice on Manifold
15	2250	60	33.5	14668	978	Ice on Manifold
16	2250	60	31	16052	1003	Ice on Manifold, sidewall and bottom
50.5	2750	133	56	54631	1082	No Flooding
50.5	2750	134	56	55341	1095	No Flooding
51.0	2750	134	55	56050	1099	No Flooding

5.0 COLD TRAP

A cryogenic cold trap was designed and fabricated during the program to provide NASA-JSC a piece of test equipment to permit testing of a flash evaporator system at the Building 7 facilities. The current vacuum chamber facility, a 1.83 m by 3.048 m chamber, does not have a LN₂ cooled shrouding which is required to condense out water from the evaporator prior to entering the vacuum pumping facility.

Design Requirements

The cold trap must interface with the NASA vacuum pumping facility and the operational chamber containing the flash evaporator. The interface requirements are standard flanges: 203.2 mm (8") diameter at the downstream vacuum pumping facility interface, and 304.8 mm (12") diameter at the vacuum test chamber.

The water condensation/pressure requirements are:

Water Condensation: 59 Kg/hr (130 lb/hr) for 1 hour
6.8 Kg/hr (15 lb/hr) continuous operation
Pressure : 1 mm Hg or less

Design Description

Figure 5-1 shows the anticipated installation of the cold trap at NASA-JSC identifying the interfaces and the overall configuration of the device. The LN₂ cryo-pumping surfaces are contained within the two 406.4 mm (16") diameter by 3.6 m (12 ft.) long aluminum tube sections to provide the desired water condensation performance. The two separate sections are used so that while one section is operating, the other section can be thawed. This will provide for continuous operation. Two butterfly valves are used to isolate each section so that the section needing thawing can be warmed and subsequently drained while the other section is still operating.

The cold trap sections are connected to the vacuum test chamber by standard 304.8 mm (12") diameter schedule 40 aluminum tubing with the final connection to the chamber made through a standard 150 lb. flange. This diameter connecting tubing is required to provide adequate cross-sectional area to prevent choked flow at maximum water vapor flow conditions.

The cold trap sections are connected to the vacuum pumping facility by standard 203.2 mm (8") diameter schedule aluminum tubing with the final connection made to the NASA interface with a standard 150 lb. flange.

The cold trap LN₂ flow schematic is shown in Figure 5-2. Liquid nitrogen is provided by the NASA facility through an 25.4 mm (1") diameter connection. From there it flows through a flow regulating valve into the cold trap surfaces. The cold trap surfaces are formed from the aluminum extrusions shown in Figure 5-3. The LN₂ makes four passes through the 3.657 m (12') section, exits the cold trap section, passes through a back pressure control

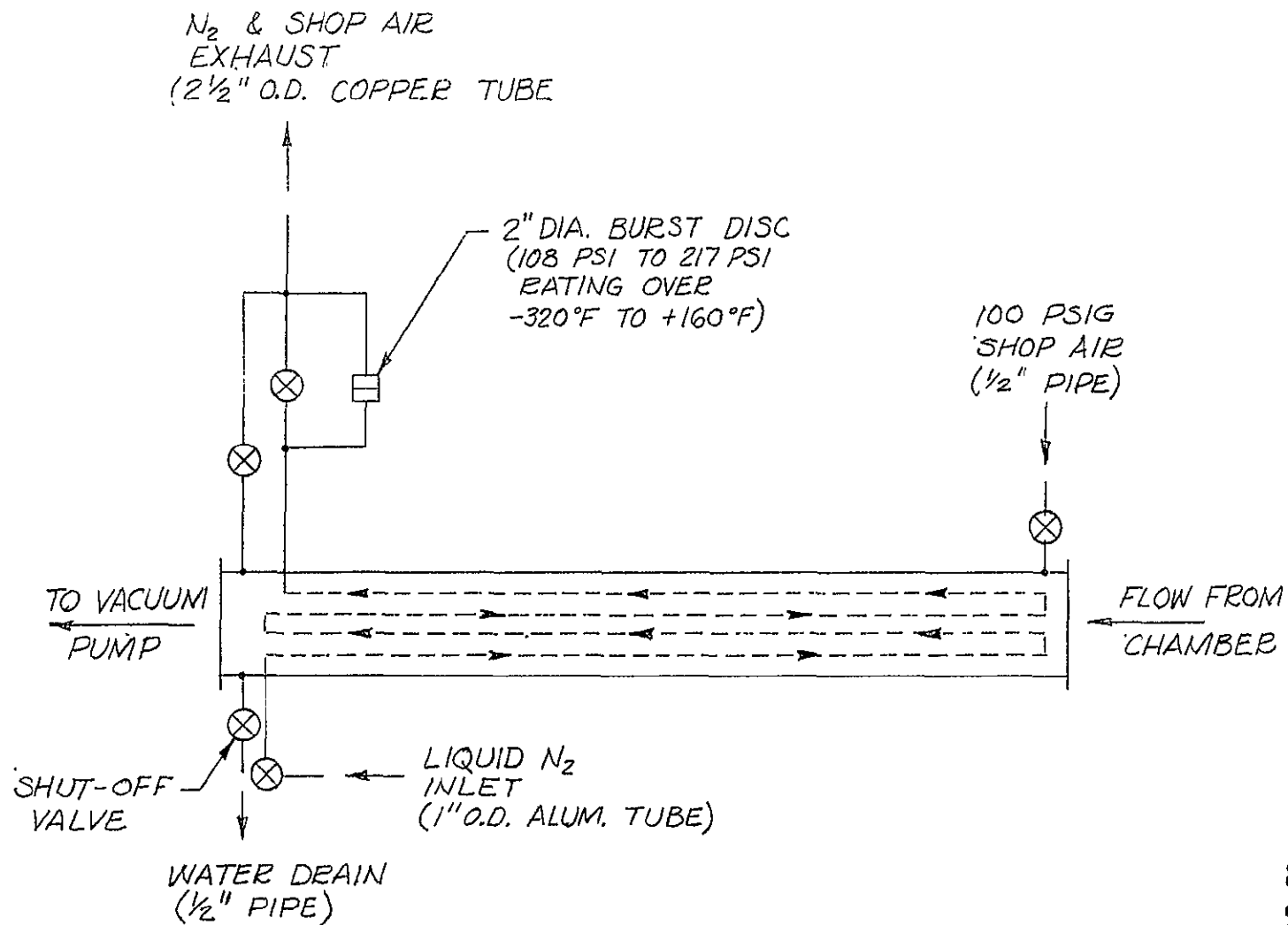


FIGURE 5-2. COLD TRAP SECTION SCHEMATIC

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valve, and flow into a NASA exhaust line. A burst disc is put in parallel with the back pressure control valve as a safety device to permit pressure relief of trapped LN₂ in the case both the back pressure control and LN₂ regulating valve are both closed.

In each cold trap section, forty aluminum extrusions of the cross-section shown in Figure 5-3 form the cold plate surfaces to condense the flash evaporator water. The extrusion assembly provides 211 ft² of surface area for water condensation per cold trap section. This will result in an ice thickness of less than 1.5 mm (0.06 in.) when both sections of the cold trap are operated at the required maximum heat load for the one hour period will build up on the cold plates. The cold trap extrusions are flow connected in a combination of series and parallel arrangements to minimize pressure drop. At the end of each pass down the length of the extrusion, the tubes are manifolded together prior to making the next pass. This provides a constant quality LN₂ - gaseous N₂ mixture for each tube flowed in parallel.

The cold plate surfaces are structurally attached but thermally isolated from the 406.4 mm (16") diameter containment tubes at the downstream end so that there are minimal stresses between the cold plate section. At the upstream end, the extrusion assembly is not attached to the containment tubes and are allowed free movement in them. This provides the capability to accommodate differential thermal expansion and contraction between the extrusion assembly and tubes.

Thawing or melting of the condensed flash evaporator water is accomplished (after LN₂ flow is stopped and the section has been isolated from the vacuum sources by the butterfly valves) by flowing warm shop air over the iced cold trap surfaces. A 12.7 mm (1/2") connection to NASA 695 Ka (100 psi) shop air is provided (see Figure 5-2). The air passes through a shut-off valve, flows externally over the cold plate extrusion on which ice has deposited, exits the cold trap section through another shut-off valve, and flows to the 63.5 mm (2-1/2") NASA exhaust system. The ice is melted by the warm air, and the water flows to the downstream end of the cold trap section by gravity due to the 63.5 mm (2-1/2") tilt of the section. The water then flows through a 12.7 mm (1/2") water drain to the NASA facility.

At the completion of the thawing, the cold trap section can be activated by: a) closing the shop air valve, the air exhaust valve, and the water drain valve; b) initiating LN₂ flow to the cold trap; and c) opening the butterfly isolation valves.

Fabrication and Checkout

The cold trap design described above was fabricated and is shown in Figure 5-4. The cold trap sections were pressure checked and shock tested by flowing LN₂ through the device to assure design integrity.

Installation and checkout testing are to be performed at NASA/JSC subsequent to the release of this report. The checkout testing will verify the adequacy of the design to meet the performance requirements. The RSEHS test article described in Section 4.0 will provide the water vapor for the test.

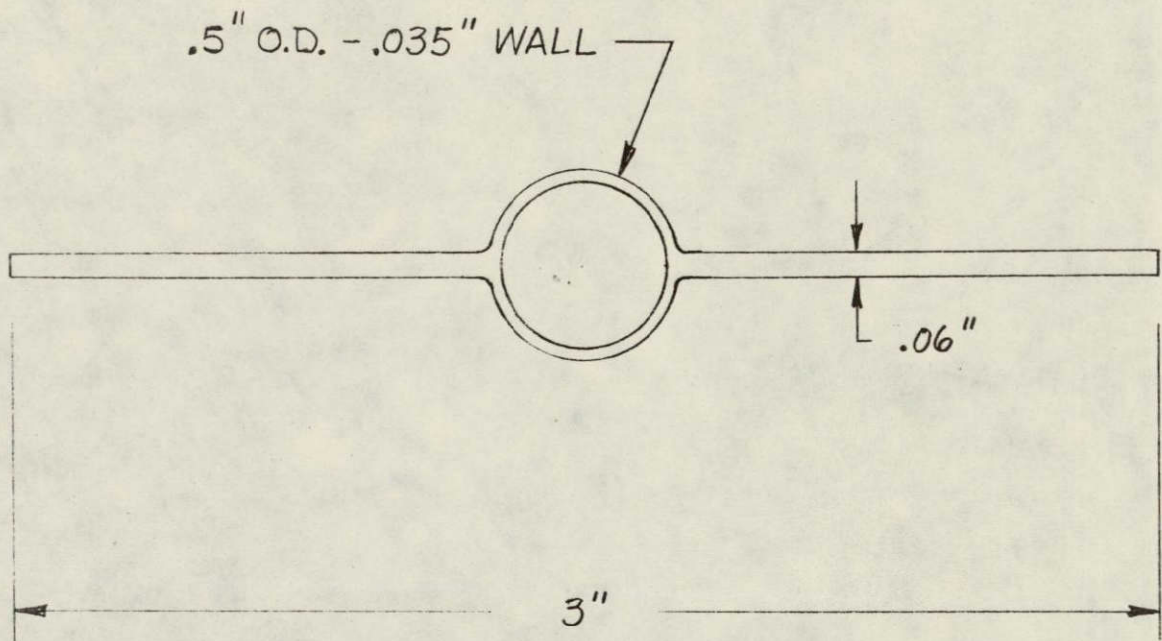


FIGURE 5-3 COLD TRAP ALUMINUM EXTRUSIONS

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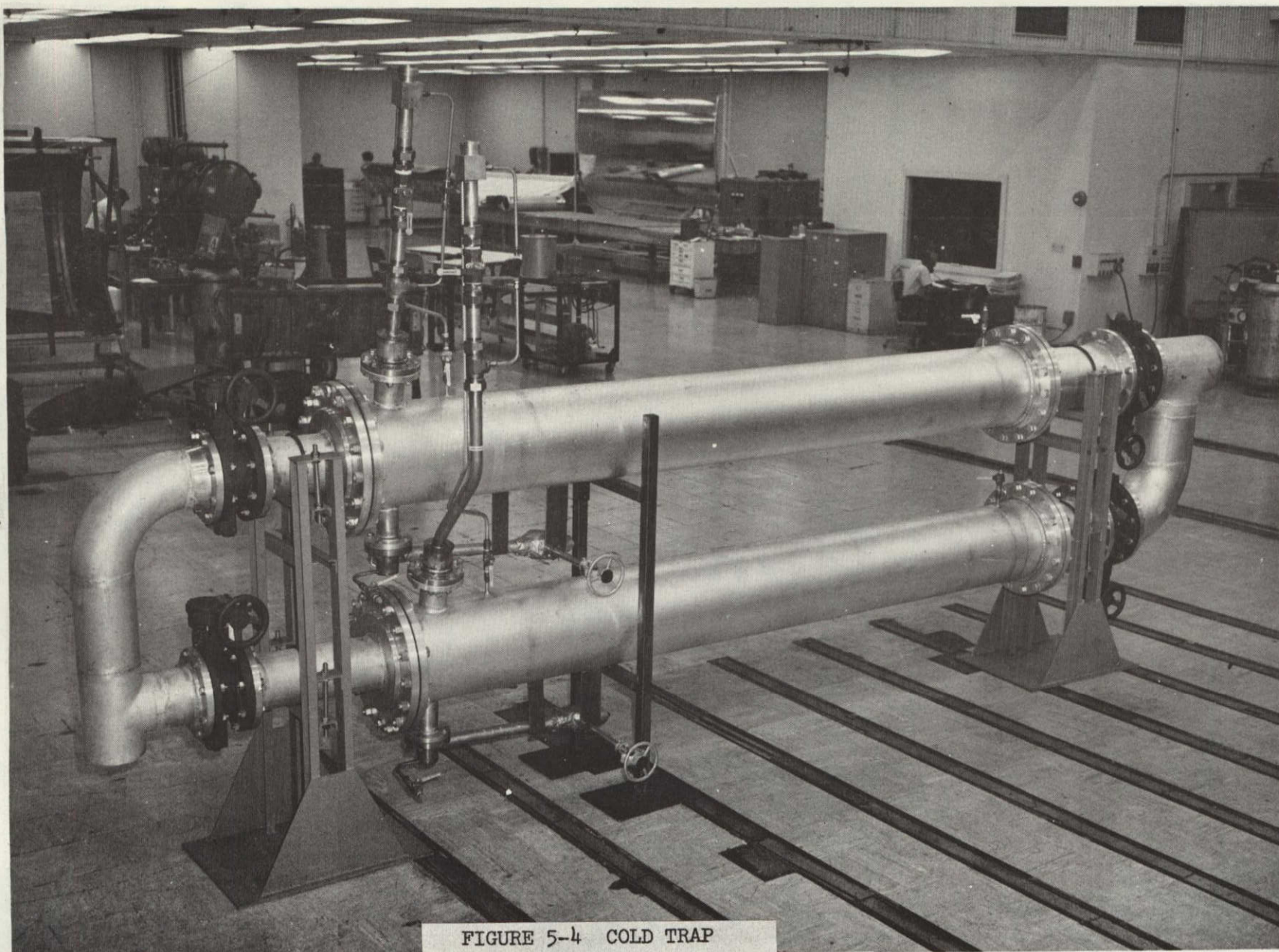


FIGURE 5-4 COLD TRAP

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6.0 REFERENCES

- (1) Hixon, C. W., and Dietz, J. B., "Development and Parametric Evaluation of the Prototype 2 and 3 Flash Evaporators", Vought Corporation, Report No. T157-76, 18 July 1975.
- (2) Dietz, J. B., "Flash Evaporator Systems Test, Final Report", Vought Corporation, Report No. 2-53230/6R-3353, 23 December 1976.